PROPELLERS

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PREFACE

THIS book gives a reliable and convenient method of designing propellers, based on model experiments, and free from theoretical intricacies and uncertainties. For details of experiments on which the method is based and for theoretical investigations, reference may be had to the author's *Naval Architecture.*

Tables are given for two, three, and four bladed propellers, from which the dimensions of a propeller may readily be determined, without interpolation, when the power, speed and revolutions are assigned, either to give maximum efficiency or to conform to certain restrictions such as limited draught.

Drawings and computations for propellers are much simplified by using the projected contour and area of the blades, and tables are given by aid of which exact results may quickly be determined. The designer may, however, use the conventional developed contour of blades, if he prefers; simple methods of drawing such contours are included in the text.

A brief treatment is given of methods for determining the power required to propel a ship at a given speed, together with data for various types of ships and boats. It is believed that intelligent use of this material will give satisfactory results under ordinary conditions; the best results, expecially under extreme conditions, can be expected only by experienced designers who have specific information.

All methods of designing ships and propellers are based, either explicitly or implicitly, on the theory of mechanical similitude. The conclusions and methods of this theory may be stated briefly and used easily; a presentation of the theory is given at the end of the book for convenient reference.

PROPELLERS

In the design of a propeller the first thing is the determination of the power required to drive the ship at the desired speed. This is at once evident when the power is underestimated because the engine cannot turn the propeller up to the designed number of revolutions, and so cannot develop its power. A moderate overestimate of power may merely result in a somewhat higher speed provided the engine can stand a moderate increase above its normal speed. But if the power is much overestimated the engine will tend to run at a dangerous speed and when restrained (as by throttling the steam) will be unable to develop its power and therefore fail to give the designed speed. In either case a new design for the propeller must be made suitable to the speed at which the engine can drive the ship. For untried conditions designers commonly estimate power liberally, and a moderate excess over estimated speed is considered to be a triumph; but it is at the expense of a costly engine and a reduced carrying capacity.

Methods of Estimating Power.—There are four recognized methods of estimating power for a ship:

(1) The Admiralty coefficient.
(2) The law of comparison.
(3) Independent estimate.
(4) Model experiments.

The first two methods are direct applications of the theory of similitude and the other two employ the conclusions of that theory with modifications. The use of the methods will be

illustrated; a discussion of the theory will be found at the end of the book.

Admiralty Coefficient.—One of the best known, and most convenient methods of estimating power, for a ship is by aid of the equation,

$$\text{I.H.P.} = \frac{1}{K} D^{\frac{2}{3}} V^3, \quad \ldots \ldots \quad (1)$$

in which

I.H.P. is the indicated horse-power of the steam-engine;

D is the displacement in tons (2240 pounds);

V is the speed in knots (6080 feet) per hour;

K is a numerical coefficient known as the Admiralty coefficient.

This coefficient must be determined from some ship for which the displacement, power, and speed are known. Values for certain ships of various types are given in the table on pages 8 and 9. The value of the coefficient may vary from 100 to 300; it is therefore evident that the success of this method depends on a judicious selection of the coefficient. In order that a close concordance may be expected between the estimated speed and the actual speed on trial of the ship, the coefficient must be derived from a ship that is geometrically similar to the ship under design, and which has the corresponding speed. These terms will be explained on page 4. A moderate deviation from these conditions will not seriously affect the value of the method.

For turbine steamers and for ships and boats which are driven by internal-combustion engines the shaft horse-power is given instead of the indicated horse-power, and the equation may read

$$\text{S.H.P.} = \frac{1}{K} D^{\frac{2}{3}} V^3. \quad \ldots \ldots \quad (2)$$

The coefficient must in such case be taken from information concerning ships or boats with turbines or internal-combustion engines, more especially because the types of propellers are different. If information is lacking the indicated horse-power may be computed by equation (1) and the shaft horse-power may be estimated

by multiplying by a factor which varies from 0.85 to 0.90; but this method must be used cautiously.

Example.—Let it be required to determine the power for a ship to have 28,600 tons displacement and a speed of 25 knots per hour.

The express steamship *Campania* in the table on page 8 has 276 for the Admiralty coefficient, and using equation (1) we find for the power,

$$\text{I.H.P.} = \frac{\overline{28600}^{\frac{2}{3}} \times \overline{25}^{3}}{276} = 52900.$$

In the solution of this equation it is convenient to take the two-thirds power of the displacement from the table on page 123, and the cube of the speed from the table on page 125, interpolating when necessary. This gives

$$\text{I.H.P.} = \frac{935 \times 15625}{276} = 52900,$$

the numerical work being most readily done by aid of a slide rule.

If preferred, the computation can be made by aid of logarithms; the parallel computations give a valuable check.

$$\begin{array}{rr}
\log 28600 = 4.4564 \\
2 \\
\hline
3)8.9128 \\
\hline
2.9709 \\
4.1937 \\
\hline
7.1646 \\
\log 276 = 2.4409 \\
\hline
\log 52900 = 4.7237
\end{array}
\qquad
\begin{array}{rr}
\log 25 = 1.3979 \\
3 \\
\hline
4.1937
\end{array}$$

Example.—Required the power for a motor boat weighing 1600 pounds to make a speed of 15 statute miles per hour. The speed in knots in this case becomes

$$\frac{15 \times 5280}{6080} = 13 \text{ knots}.$$

The displacement is

$$1600 \div 2240 = 0.714 \text{ ton};$$

The power consequently is by equation (2)

$$\text{S.H.P.} = \frac{(0.714)^{\frac{2}{3}} \times \overline{13}^3}{165} = 10.6,$$

the Admiralty coefficient being taken for *Chum*, page 9. The solution by logarithms is

$$\begin{array}{rl}
\log 0.714 = & 9.8537 - 10 \\
 & \underline{\qquad\qquad 2} \\
 & 19.7074 - 20 \\
 & \underline{10.0000 - 10} \\
3) & \underline{29.7074 - 30} \\
 & 9.9025 - 10 \\
 & \underline{3.3417} \\
 & 3.2442 \\
\log 165 = & \underline{2.2175} \\
\log 10.6 = & 1.0267
\end{array}
\qquad
\begin{array}{r}
\log 13 = 1.1139 \\
\underline{3} \\
3.3417
\end{array}$$

Similarity.—Geometrical figures are said to be similar when they have the same form and differ only in size. A ship and its model are made from the same lines and differ only in scale; the first may be several hundred feet long and the latter only a few feet long.

Mechanical Similitude.—The theory of mechanical similitude is an extension of geometrical similitude to the conceptions of mechanics including force, work, and power. For the present purpose the applications of the theory to the design of a ship and its propeller will be stated; a simple presentation of the theory is given at the end of the book for convenient reference.

Corresponding Speeds.—*The corresponding speeds for similar ships are proportional to the square roots of their lengths.*

Example.—The *Campania* has a length of 600 feet and makes 23.18 knots per hour (page 8); a ship 700 feet long on the same lines would have a corresponding speed of

$$\sqrt{600} : \sqrt{700} :: 23.18 : V, \quad \therefore V = 25 \text{ knots.}$$

Example.—If a ship 700 feet long makes 25 knots per hour, then the corresponding speed for a model 20 feet long will be

$$\sqrt{700} : \sqrt{20} :: 25 : V_m, \quad \therefore V_m = 4.23 \text{ knots,}$$

which is the speed at which such a model should be towed in an experimental towing basin in order to investigate the relative powers of the ship and its model.

Example.—Conversely if a ship 600 feet long makes 23.18 knots per hour, then a ship to make 25 knots must have a length of

$$\overline{23.18}^2 : \overline{25}^2 :: 600 : L, \quad \therefore L = 700 \text{ (nearly),}$$

provided that the conditions of the theory of mechanical similitude are observed.

Displacement.—The displacement of a ship is given in tons (2240 pounds); small yachts and boats may have displacements given in pounds; this custom is commonly applied to craft that can conveniently be weighed complete.

The volume of water displaced by a ship when afloat can be determined from the lines and is stated in cubic feet; this may be called the volumetric displacement.

Now 35 cubic feet of sea-water weigh one ton; for fresh water it is customary to take 36 cubic feet to the ton. The displacement of a ship in tons is obtained by dividing the volumetric displacement in sea-water by 35; for fresh water divide by 36.

Similar ships have displacements proportional to the cubes of their lengths.

Example.—The *Campania* has a displacement of 18,000 tons on a length of 600 feet; a similar ship 700 feet long will have a displacement of

$$\overline{600}^3 : \overline{700}^3 :: 18000 : D, \quad \therefore D = 28600 \text{ tons.}$$

Example.—If a ship 700 feet long has a displacement of 28,600 tons, then a model 20 feet long will weigh

$$\overline{700}^3 : \overline{20}^3 :: 28600 : D_m, \quad \therefore D_m = 0.667 \text{ ton},$$

or 1494 pounds.

Dimensions.—The main dimensions of a ship are length, beam, and draught.

The length is measured between *perpendiculars* drawn at the *load-water-line.* The forward perpendicular is drawn at the forward side of the stem, and the after perpendicular is drawn at the after side of the stern-post.

The beam is the extreme beam. This is usually found at the load-water-line nearly half way between perpendiculars.

The draught is measured from the water-line to the bottom of the keel, midway between perpendiculars. If the ship has no external keel the draught is measured to the outside of the keel-plate. The extreme draught is frequently given also if it is different from the draught at mid-length, but it does not enter into computations with the main dimensions.

Load-water-line.—A ship is designed for a certain normal displacement. The side-elevation or *sheer-plan* has a line drawn on it showing the draught at this normal displacement; this line is the load-water-line. The surface of the water when the ship is afloat at the normal displacement will be at the height of the normal draught; the plane of the surface of the water is the plane of the load-water-line; it is frequently referred to briefly as the load-water-line.

At other than the designed normal displacement the ship will float at some other water-line. Any line or plane parallel to the surface of the water may be called a water-line.

Problem.—So far we have considered only simple examples relating to power for a ship; in general there are various conditions and restrictions on the design for a ship that will call for consideration before the Admiralty coefficient can be chosen.

Let it be required to determine the power for a ship 700 feet long, and having a displacement of 28,600 tons, to make 25 knots per hour.

In order to decide whether the power for such a ship can be determined from the power of the *Campania* we may make the following comparison. That ship has a length of 600 feet, a beam of $65\frac{1}{4}$ feet, and a normal draught of 25 feet; at a displacement of 18,000 tons the speed is 23.18 knots per hour.

A similar ship 700 feet long will have the dimensions,

$$600 : 700 :: 65\tfrac{1}{4} : B, \quad \therefore \text{beam} = 76.1 \text{ feet.}$$

$$600 : 700 :: 25 \;\; : d, \quad \therefore \text{draught} = 29.2 \text{ feet.}$$

The displacement will be

$$\overline{600}^3 : \overline{700}^3 :: 18000 : D, \quad \therefore D = 28580 \text{ tons.}$$

The corresponding speed will be

$$\sqrt{600} : \sqrt{700} :: 23.18 : V, \quad \therefore V = 25.0 \text{ knots.}$$

If the beam and draught can be accepted then the design can be based properly on the data for the *Campania;* in practice the draught for large ships is commonly restricted by the depth of channel in harbors. Should a restriction to a draught of 28 feet be imposed in this case then some change from similarity would be required. We might (1) increase the beam, (2) increase the length, or (3) use fuller lines; or any two or all three conditions might be changed. It is but fair to say that the problem was made up from the data of the *Campania* with slight modifications from the conditions for similarity and that so close concordance cannot generally be expected. Since the method of the Admiralty coefficient has some flexibility a good determination of power can be had by use of coefficients from ships that are not very dissimilar.

To complete the problem the computation on page 3 will be transferred, giving

$$\text{I.H.P.} = \frac{\overline{28600}^{\frac{2}{3}} \times \overline{25}^3}{276} = 52900.$$

Type Ships.—Successful use of the Admiralty coefficient (or of the theory of similitude either directly or with modifications)

DATA FOR VARIOUS SHIPS.

Type.	Name.	Date.	Length B.P.	Beam.	Trial Draft.	Trial Disp. Tons.	Block Coef. at Trial Disp.	Wetted Surface. Sq. Ft.	Speed, Knots.	I.H.P.	I.H.P. per Ton Disp.	Adm. Coef.	Pris. Coef.	R.P.M. Trial.	$\frac{V}{\sqrt{L}}$
High-speed passengers	Lusitania......	1908	760–0	87–6	32–9	37,080	.596	85,500	25.62	*76,000	2.05	245	......	194	.93
	Deutschland ...	1900	662–9	67–0	29–0	23,200	.63	64,100	23.50	35,500	1.53	297	......	76	.91
	Campania	1893	600–0	65–3	25–0	18,000	.64	49,620	23.18	31,050	1.72	276	......	......	.95
	La Provence...	1906	597–0	64–7	26–8	19,160	.649	54,500	22.05	30,000	1.66	255	......	......	.90
Intermediate passenger and freight	Otaki[1]........	1909	464–6	60–0	20–1	11,716	.735	35,700	15.02	6,857	.58	255	.....	103–224	.70
	Saxonia[2]	1900	580–0	64–2	29–0	22,580	.732	60,800	16.00	10,150	.45	323	......	......	.66
	Laos..........	1899	442–0	50–10	24–4	8,910	.570	34,100	18.50	9,000	1.00	302	......	......	.88
	Minnesota[4]	1903	608–0	73–0	33–0	33,000	.790	75,200	14.00	10,000	.303	282	.80	......	.57
Coasters	Pawnee........	1907	255–9	40–0	15–6	2,840	.650	13,072	11.5	1,170	.41	267	.724	111	.72
	Howard.......	1902	272–0	42–0	16–6	3,224	.600	14,684	13.0	1,770	.548	270	......	66	.79
	J. S. Whitney...	1901	272–0	43–0	13–11	2,792	.60	14,027	14.5	2,230	.80	271	......	......	.88
Channel str..	Alberta........	1900	270–0	35–6	10–10	1,534	.518	10,250	19.9	5,350	3.49	196	......	......	1.21
Sound str....	City of Lowell. .	1894	320–0	48–0	12–10	2,445	.434	13,855	19.2	4,347	1.92	296	.573	126	1.07
Freighters and tramps	Nebraskan.....	1902	358–2	46–0	23–3½	8,380	.76	28,990	10.3	1,635	.195	262	.79	69.5	.55
	Pennsylvania...	......	430–0	50–0	17–4	8,926	.83	30,520	9.89	1,455	.163	286	......	......	.48
	Cargo str. L.[2] ..	1900	420–0	53–2	23–9½	11,940	.788	36,100	10.84	2,494	.209	267	......	......	.53
	Delaware......	......	345–0	44–0	24–6	8,200	.77	26,490	12.27	2,686	.327	279	......	......	.66
Yachts....	Noma.........	1903	226–0	28–5	13–0	1,035	.434	8,000	19.06	4,200	4.06	163	.601	205	1.27
	Vanadis.......	1908	232–8	32–6	12–6	1,467	.544	9,540	13.00	1,050	.716	270	......	152	.85
	Tarantula[3]	1902	152–6	15–3	5–0	145	.437	....	23.36	2,200	15.16	201	......	1200	1.89
	Lorena[3]	1903	253–0	33–3	13–0	1,400	.448	9,500	18.02	*3,800	2.71	193	.....	550–700	1.13

Side-wheelers	Commonwealth	1908	437–11	55–0	13–0½	5,430	.608	25,200	20.05	12,000	2.21	209	.626	29.8	.96
	Tashmoo......	1901	300–0	37–6	8–3	1,224	.463	8,976	18.85	3,400	2.79	224	......	40	1.09
	Uncatena......	1903	179–4	31–4	6–3	608	.603	5,535	13.0	982	1.63	160	......	29	.97
	City of Erie....	1901	314–0	44–0	10–3	2,233	.550	12,776	19.1	6,450	2.90	185	......	33	1.08
Battle-ships	Louisiana......	1906	450–0	76–5½	24–6	15,923	.661	44,500	18.82	20,442	1.28	206	.687	127	.89
	Rhode Island...	1906	435–0	76–2½	23–9	14,680	.652	42,000	19.01	19,889	1.36	207	.690	125	.91
	North Carolina	1908	502–0	72–6	25–0½	14,540	.560	44,700	21.92	26,038	1.79	242	......	118	.98
	Delaware......	1909	510–0	84–10	27–0	20,098	.603	53,000	21.56	28,578	1.43	259	......	128	.95
Fast cruisers	Tennessee.	1906	502–0	72–11	25–0	14,500	.555	43,450	22.16	26,540	1.83	244	.585	127	.99
	Montana	1908	502–0	72–6	25–0	14,531	.558	43,500	22.26	27,489	1.89	239	......	123	.99
Scouts	Birmingham ...	1908	420–0	46–8	16–7½	3,722	.410	19,900	24.33	15,476	4.16	223	.556	191	1.18
	H.M.S.Forward	1905	365–0	39–2	13–10½	2,790	.492	....	25.25	14,990	5.38	213	......	......	1.32
Destroyers	Whipple.......	1901	248–0	22–6	8–5	487	.363	....	28.05	6,766	13.9	202	......	309	1.78
	Flusser........	1910	289–0	26–5	7–11½	686	.405	7,150	30.40	11,541	16.8	190	.627	800	1.76
	H.M.S.Cossack.	1909	270–0	26–0	8–2	836	.510	....	33.00	*18,000	21.5	177	......	670	2.01
Gunboat....	Dubuque......	1905	174–0	35–0	12–3	1,085	.509	6,900	12.90	1,180	1.09	192	.627	228	.98
Revenue....	Manning	1899	188–0	32–0	12–4	1,000	.471	7,273	16.00	2,181	2.18	183	.605	152	1.17
High-speed motor-boats	Dixie II.......	1908	39–3	4–8	0–11	2.01	.420	....	31.05	*220	109.5	217	......	900	4.95
	Legru-Hotchkiss....	1904	39–11	5–0	1.0	2.26	.360	....	29.60	*170	75.2	262	......	780	4.70
	Wolsey-Siddeley	1908	39–4	6–0	......	4.40	......	....	27.35	*414	94.2	133	......	1000	4.36
	Quicksilver	1904	30–0	5–11	0–11	1.52	.328	....	18.20	*39	25.7	205	......	900	3.33
	Chum.........	1906	25–0	4–0	0–10	.725	.304	....	12.70	*10	13.8	165	......	......	2.54

(1) Wing engines; centre turbine. (2) At sea. (3) Triple Screw Turbines. (4) Designed. * S.H.P.

will depend on the exactness and certainty of information in the hands of the designer, from ships which have been tested under known conditions.

Some builders make a practice of testing all ships built (or at least all types of ships) and they have means of estimating power for new designs with certainty and precision, unless such designs differ radically from previous ships.

There is much published information concerning power and speed of merchant ships, but analysis of such information will show that commonly the data (or part of the data) come from the design and not from trials; even when trials are made they are liable to be for ships light instead at the designed draught, or it may be that some of the conditions are not definite. It is difficult to load freight ships, or freight and passenger ships to the normal draught for trials, and trials under service conditions are often indefinite. Published data of designs have a value especially when given out by well-known designers or builders.

The results of acceptance trials of warships are freely published and conditions are frequently given completely and precisely. And yet discrepancies between results from ships of the same class leave much to be desired. Such ships are driven at high speeds at which secondary influences have large effects. This is particularly true of torpedo-boats and destroyers. Success with such craft can be expected only by experienced builders who keep complete trial data; even they occasionally meet with disappointment under new conditions.

Fast yachts and motor-boats are also driven at very high relative speeds and success demands the same conditions.

On pages 8 and 9 will be found data from various types of ships and boats, which can be used for practice or in lack of better information.

In addition to the dimensions of the ships, it is customary to give the block-coefficient, the wetted surface and the horse-power per ton of displacement. In some cases the prismatic coefficient is given, and in our table the last column gives the speed-length-ratio.

Block-coefficient.—This coefficient is the ratio of the displacement of the ship to that of a rectangular block having the same length, beam and draught. It may vary from 0.35 to 0.85.

The block-coefficient of similar ships is necessarily the same, but ships having the same block-coefficient may be dissimilar. On the whole this coefficient gives a fair idea of the effect of variations of form among ships of the same class.

Example.—The block-coefficient for the *Campania* is

$$B = \frac{18000\times 35}{600\times 65.25\times 25}=0.644. = \frac{V\ \text{cu ft}}{L' B' H'}$$

The numerator contains the displacement in tons multiplied by the volume of sea-water per ton; the denominator contains the main dimensions of the ship.

Wetted Surface.—The surface of the ship in contact with the water can be determined from the lines of the ship and is given in square feet. The necessary operations are tedious and require skill of the draughtsman.

For a preliminary design the wetted surface may be computed by the equation

$$\text{Wetted surface} = C\sqrt{DL} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

in which D is the displacement of the ship in tons, L is the length in feet and C is a coefficient to be selected from the following table where B is the beam and H is the draught.

$B \div H$	C	$B \div H$	C	$B \div H$	C
2.0	15.63	2.5	15.50	3.0	15.62
2.1	15.58	2.6	15.51	3.1	15.66
2.2	15.54	2.7	15.53	3.2	15.71
2.3	15.51	2.8	15.55	3.3	15.77
2.4	15.50	2.9	15.58	3.4	15.83

The error of this method may amount to 2.5 per cent for ships which are either very full or very fine. For merchant ships the error is usually not more than 2 per cent.

Example.—The wetted surface for the *Campania* is given on page 8 as 49,620 sq. ft. The ratio of beam to draught is

$$65.25 \div 25 = 2.6$$

for which the table gives $C = 15.51$. The displacement is 18,000 tons and the length is 600 feet; consequently the wetted surface may be computed to be

$$15.51\sqrt{18000 \times 600} = 51000 \text{ sq. ft.}$$

Law of Comparison.—The theory of mechanical similitude as applied to determining power for a ship is known as the extended law of comparison. This law is:

The horse-powers of similar ships at corresponding speeds are proportional to the seven-sixths powers of the displacements.

Problem.—Let it be required to determine the dimensions and power of a ship to make 25 knots per hour, using the *Campania* for the type ship. The *Campania* makes 23.18 knots on a length of 600 feet; the corresponding speed for the new ship will give

$$23.18 : 25 :: \sqrt{600} : \sqrt{L}, \quad \therefore \; L = 700 \text{ feet (nearly).}$$

The beam and draught as computed on page 7 are 76.1 feet and 29.2 feet, and the displacement is about 28,600 tons.

The extended law of comparison gives

$$\overline{18000}^{\frac{7}{6}} : \overline{28600}^{\frac{7}{6}} :: 31050 : \text{I.H.P.}, \quad \therefore \text{I.H.P.} = 53300.$$

This problem may conveniently be solved by logarithms as follows:

$$\begin{array}{r}
\log 28600 = 4.4564 \\
\log 18000 = 4.2553 \\
\hline
.2011 \\
7 \\
\hline
6)\underline{1.4077} \\
.2346 \\
\log 31050 = 4.4921 \\
\hline
\log 53300 = 4.7267
\end{array}$$

Should the computation be for a smaller ship the order for logarithmic work may be changed as follows. Suppose the displacement were 12,000 tons; then

$$\overline{18000}^{\frac{7}{6}} : \overline{12000}^{\frac{7}{6}} :: 31050 : \text{I.H.P.} \quad \therefore \text{I.H.P.} = 19300$$

$$\begin{array}{rr}
\log 18000 = 4.2553 & \log 31050 = 4.4921 \\
\log 12000 = 4.0792 & .2055 \\
\hline
.1761 & \log 19400 = 4.2866 \\
7 & \\
\hline
6)1.2327 & \\
\hline
.2055 &
\end{array}$$

Change of Speed.—In using the laws of similitude it will frequently happen that the desired speed will differ from that derived from the type ship. If the difference is large another type ship must be chosen especially when the speed is high. If the difference between the desired speed and the corresponding speed is small then we may allow, for the change of speed on the assumption that the power varies according to the law:

The power for a ship is proportional to the cube of the speed.

Example.—The power required to drive the *Campania* at 26 knots per hour will be approximately

$$\overline{25}^3 : \overline{26}^3 :: 31050 : \text{I.H.P.} \quad \therefore \text{I.H.P.} = 34900.$$

Change of Displacement.—A ship is designed for a certain normal displacement but frequently is loaded to a different displacement and it is important to know what influence such a change will have on the speed. This matter has no relation to the theory of similitude because the ship at a different draught will have an under-water body which is not similar to that at normal draught. In particular the relation of beam to draught and the block-coefficient will be different, and both of these features have an appreciable effect on propulsion.

In much the same way it may be found that the design for a ship is restricted in draught and cannot have the draught that the

laws of similitude would indicate, when used with the proportions of a certain type ship. Also the lines may be fuller (or finer) and the displacement may thus vary from that computed by the laws of similitude.

The best method of finding the influence of displacement on speed is by trials of ships at various draughts; such trials are seldom made. When models are tried to determine power, they are frequently towed at various draughts.

This subject is both difficult and uncertain but we may use the following equation for allowing for small changes of draught or displacement

$$(\text{I.H.P.})_1 : (\text{I.H.P.})_2 :: D_1^n : D_2^n$$

and the value of the exponent n may vary from $\frac{2}{3}$ for large ships of moderate speed to $\frac{1}{6}$ for ships and boats at high speeds.

Problem.—Let it be required to determine the dimensions and power of a ship 700 feet long and having a displacement of 28,000 tons to make 25.5 knots per hour.

First let the problem be solved directly from comparison with the *Campania* and afterwards allow for change of displacement and speed.

The relative speed of a ship 700 feet long will be found by the equation

$$\sqrt{600} : \sqrt{700} :: 23.18 : V, \quad \therefore V = 25 \text{ knots.}$$

The displacement of a ship 700 feet long and similar to the *Campania* as shown on page 5 will be 28,600 tons.

Such a ship at 25 knots per hour should have 53,300 I.H.P. as computed on page 12; at 25.5 knots the power would be

$$\overline{25}^3 : \overline{25.5}^3 :: 53300 : \text{I.H.P.}, \quad \therefore \text{I.H.P.} = 56600.$$

If the power is proportional to the two-thirds power of the displacement the design for 28,000 tons will call for

$$\overline{28600}^{\frac{2}{3}} : \overline{28000}^{\frac{2}{3}} :: 56600 : \text{I.H.P.}, \quad \therefore \text{I.H.P.} = 55800.$$

Speed-length-ratio.—The rule for corresponding speed shows that intelligent comparison of speeds of ships must take account of the lengths. For this purpose we may use the speed-length-ratio expressed by the ratio

$$\frac{V}{\sqrt{L}}$$

in which V is the speed in knots per hour and L is the length in feet. A study of the table on page 8 will show that the speed-length-ratio is approximately as follows:

	Ratio.
Freighters	0.5 to 0.55
Passenger ships	0.7 to 0.8
Fast passenger ships	0.9 to 1.0
Battleships	0.9 to 1.0
Cruisers	1.0 to 1.2
Torpedo-boats and destroyers	1.8 to 2.0
Fast motor boats	2.5 to 5.0

In a rough way all craft having a speed-length-ratio under unity may be classed as slow or moderate speed, and all with a greater ratio, as fast.

Model Basins.—In order to understand the methods of estimating power which are called independent estimate and model experiments it is necessary to know how model experiments are made and how the results are used.

Model experiments are habitually and desirably made at model basins or tanks; improvised methods in open water are difficult and liable to be misleading. Such experiment stations have costly and delicate apparatus, and experimenters must have experience and discretion to get valuable results. But the fundamental conceptions are simple.

A model basin or tank is a canal 300 or 400 feet long, about 30 feet wide and 10 feet deep. The side walls of the canal carry rails bedded on masonry. A carriage, like a traveling-crane, spans the canal and travels on the rails. This carriage is driven electri-

cally much like a trolley car and can be started quickly and driven at a uniform speed.

A model of the ship, 10 to 20 feet long, is cut to the lines of the ship and is ballasted to float with the proper displacement and trim. The model is towed from the carriage at various speeds and the *resistance* or pull on the towing apparatus is measured. This is known as the *tow-rope resistance.*

The first experiments of this sort were made by William Froude, who also determined surface friction and proposed the method of independent estimate.

Resistance.—The force necessary to maintain a ship at uniform speed is known as the resistance. When a ship is propelled by its own machinery the resistance is affected by the methods of propulsion and usually is greater than the tow-rope resistance.

As proposed by Froude, the two-rope resistance is separated into surface or frictional resistance and residual resistance. The residual resistance is further separated into wave-making resistance, eddy-making resistance and steam-line resistance.

Frictional Resistance.—It is customary to calculate the surface or frictional resistance by the equation

$$R_f = fSV^n, \qquad \ldots\ldots\ldots \quad (4)$$

in which R_f is the force, in pounds, required to overcome the surface resistance, S is the wetted surface in square feet and V is the speed in knots per hour; f and n are quantities taken from tables given on pages 17 and 18.

This equation is seldom used directly in practice but is used in building up the method of independent estimate of power.

The first two tables were derived by Naval Constructor D. W. Taylor from values published by R. E. Froude. The third table is slightly modified and extended and used by Wm. Denny and Bros. Tideman's table was derived by him from Wm. Froude's experiments.

FROUDE'S SURFACE FRICTION CONSTANTS.

Given by Taylor.

SURFACE-FRICTION CONSTANTS FOR PARAFFIN MODELS IN FRESH WATER. EXPONENT $n = 1.94$.

Length, Feet.	Coefficient.	Length, Feet.	Coefficient.	Length, Feet.	Coefficient.
2.0	0.01176	10.0	0.00937	14.0	0.00883
3.0	0.01123	10.5	0.00928	14.5	0.00887
4.0	0.01083	11.0	0.00920	15.0	0.00873
5.0	0.01050	11.5	0.00914	16.0	0.00864
6.0	0.01022	12.0	0.00908	17.0	0.00855
7.0	0.00997	12.5	0.00901	18.0	0.00847
8.0	0.00973	13.0	0.00895	19.0	0.00840
9.0	0.00953	13.5	0.00889	20.0	0.00834

SURFACE-FRICTION CONSTANTS FOR PAINTED SHIPS IN SEA-WATER. EXPONENT $n = 1.825$.

Length, Feet.	Coefficient.	Length, Feet.	Coefficient.	Length, Feet.	Coefficient.
8	0.01197	40	0.00981	180	0.00904
9	0.01177	45	0.00971	200	0.00902
10	0.01161	50	0.00963	250	0.00897
12	0.01131	60	0.00950	300	0.00892
14	0.01106	70	0.00940	350	0.00889
16	0.01086	80	0.00933	400	0.00886
18	0.01069	90	0.00928	450	0.00883
20	0.01055	100	0.00923	500	0.00880
25	0.01029	120	0.00916	550	0.00877
30	0.01010	140	0.00911	600	0.00874
35	0.00993	160	0.00907		

Given by Denny.

SURFACE-FRICTION CONSTANTS. EXPONENT, 1.825.

Length, Feet.	Coefficient.	Length, Feet.	Coefficient.	Length, Feet.	Coefficient.
40	0.00996	260	0.00870	550	0.00853
60	0.00957	280	0.00868	600	0.00850
80	0.00933	300	0.00866	650	0.00848
100	0.00917	320	0.00864	700	0.00847
120	0.00905	340	0.00863	750	0.00846
140	0.00896	360	0.00862	800	0.00844
160	0.00889	380	0.00861	850	0.00842
180	0.00884	400	0.00860	900	0.00841
200	0.00879	420	0.00859	950	0.00840
220	0.00876	450	0.00858	1000	0.00839
240	0.00872	500	0.00855		

TIDEMAN'S SURFACE-FRICTION CONSTANTS.

Derived from Froude's Experiments.

SURFACE-FRICTION CONSTANTS FOR SHIPS IN SALT WATER OF 1.026 DENSITY.

Length of Ship in Feet.	Iron Bottom Clean and Well Painted.		Copper or Zinc Sheathed.			
			Sheathing Smooth and in Good Condition.		Sheathing Rough and in Bad Condition.	
	f	n	f	n	f	n
10	0.01124	1.8530	0.01000	1.9175	0.01400	1.8700
20	0.01075	1.8490	0.00990	1.9000	0.01350	1.8610
30	0.01018	1.8440	0.00903	1.8650	0.01310	1.8530
40	0.00998	1.8397	0.00978	1.8400	0.01275	1.8470
50	0.00991	1.8357	0.00976	1.8300	0.01250	1.8430
100	0.00970	1.8290	0.00966	1.8270	0.01200	1.8430
150	0.00957	1.8290	0.00953	1.8270	0.01183	1.8430
200	0.00944	1.8290	0.00943	1.8270	0.01170	1.8430
250	0.00933	1.8290	0.00936	1.8270	0.01160	1.8430
300	0.00923	1.8290	0.00930	1.8270	0.01152	1.8430
350	0.00916	1.8290	0.00927	1.8270	0.01145	1.8430
400	0.00910	1.8290	0.00926	1.8270	0.01140	1.8430
450	0.00906	1.8290	0.00926	1.8270	0.01137	1.8430
500	0.00904	1.8290	0.00926	1.8270	0.01136	1.8430

Residual Resistance.—The residual resistance is computed from trials on ships or experiments on models, by subtracting the surface or frictional resistance from the tow-rope resistance. A convenient form for expressing residual resistance is

$$R_w = \frac{bD^{\frac{2}{3}}V^4}{L} \quad . \ . \ . \ . \ . \ . \ . \ . \quad (5)$$

where D, V, and L are the displacement in tons, the speed in knots and the length in feet, and b is a numerical factor.

Long fine ships, like Atlantic liners may have $b=0.35$; moderately fine ships may have $b=0.40$; ships broad in proportion to length but fine at ends, like war-ships, may have $b=0.45$; freight ships may have $b=0.45$ to 0.5. The value of b is also likely to be affected by speed especially when the speed-length-ratio is high.

The residual resistance for ships that have small external

appendages is mainly wave-making resistance and is frequently called by that name. It probably follows the laws of mechanical similitude (at least approximately) and may be used with fair confidence when properly derived from tests or experiments. For ships having a speed-length-ratio less than unity the wave-making resistance is not large (relatively) and may be used as a valuable check on other methods even though the factor b is uncertain.

The residual resistance is seldom used in practice, but forms an element of the method of independent estimate of power; all the reservations for residual resistance apply to that elment of the method of independent estimate.

Stream-line Resistance.—The passage of a ship through the water deflects it to the sides and it closes in again astern of the ship. This action is accompanied by the formation of a system of waves which travel along with the ship. The crests of the waves may be broken especially near the bow of the ship; but on the whole the water appears to flow past the ship in an unbroken stream. The curved path followed by a drop of water in the stream, is known as a stream line. The hydrostatic pressure of water in a stream line varies much as it would in a pipe through which water is flowing, decreasing as the velocity increases and vice-versa. There is therefore a variation in pressure along the side of the ship. If on the whole the variation of pressure of the whole stream of water which appears to flow past the ship gives an unbalanced resultant pressure, then there is stream-line resistance.

Both theory and experiment lead us to think that stream-line resistance is small for a well formed ship. In practice no attempt is made to compute stream-line resistance separately. Care should be taken that bilge-keels and other external appendages do not interfere with stream-line flow, and cause undue resistances from formation of eddies or otherwise.

Stream Lines about Ships.—To give an idea of forms of stream lines past the hulls of ships Figs. 1 and 2 are given on page 20. The first represents a cruiser with a block-coefficient of 0.53 and a speed-length-ratio of 1.1, while the second represents a collier with a speed-length-ratio of 0.7 and a block-coefficient of 0.72.

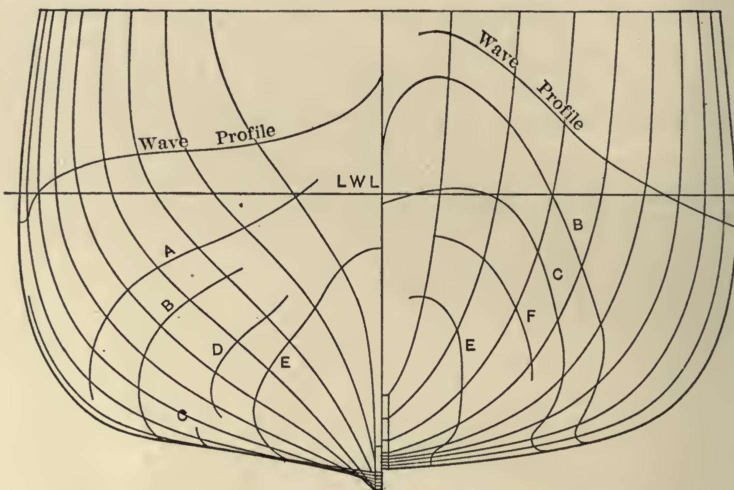


FIG. 1.—Stream Lines about a Cruiser.

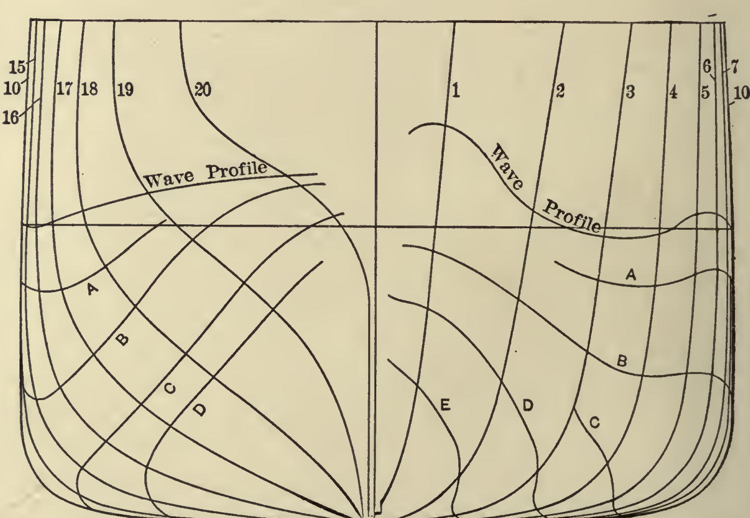


FIG. 2.—Stream Lines about a Collier.

Eddy-making Resistance.—A well formed ship of proper proportions has little if any eddy-making resistance, unless it has external appendages, like propeller-shaft struts, or spectacle-frames. Bilge-keels if they cut across stream lines and especially if extended toward the ends of the ship may cause large eddy-making resistances. Well arranged bilge-keels may give a resistance equal to two or three per cent of the resistance without bilge-keels; this is little more than the resistance computed by Froude's method for their surface.

Merchant ships with two or more shafts, usually have the propeller shafts carried by spectacle-frames. With outward turning screws the fins for such frames should droop at an angle of about $22\frac{1}{2}°$. So arranged the resistance may be 2 or 3 per cent of the resistance of the bare hull. At improper angles the resistance of such fins may be 10 or 12 per cent.

War-ships and yachts commonly have the propeller shafts carried by brackets which may increase the resistance as much as 10 per cent. The resistances of such appendages are habitually investigated at model basins when precision is desired.

Wave-making Resistance.—It has been stated of stream-line resistance and of eddy-making resistance that they individually are small for well formed ships, consequently the residual resistance can be charged mainly if not entirely to wave-making resistance. From theoretical considerations it can readily be shown that the power required to maintain the system of waves which travels along with a ship at high speed is large enough to account for most if not all the residual resistance but a useful quantitative value cannot be assigned to residual resistance in this way. It is customary to derive the form for calculating this resistance from theoretical considerations but to base the computation on comparison with tests on ships and experiments are made as previously stated.

Total Resistance.—Summing up the surface resistance as expressed by equation (4), page 16 and the residual resistance as given by equation (5), page 18, we have for the total tow-rope resistance in pounds

$$R=R_f+R_w=fSV^n+b\frac{D^{\frac{2}{3}}}{L}V^4 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

To repeat, V is the speed of the ship in knots per hour, D is the displacement in tons, S is the wetted surface in square feet, and L is the length in feet; f, n, and b are factors for which values are given on pages 17 and 18.

Independent Estimate.—Now one knot per hour is

$$6080 \div 60 = 101.3$$

feet per minute; consequently the work required to tow a ship can be found by multiplying the resistance as given by equation (6) by 101.3 V, where V is the speed in knots per hour. To find the horse-power, we may divide the work so computed by 33,000. Consequently the horse-power required to tow the ship is

$$\text{E.H.P.} = \frac{101.3}{33000} RV.$$

Replacing R by its value in equation (6) we have for the net horse-power.

$$\text{E.H.P.} = 0.00307\left(fSV^{n+1} + b\frac{D^{\frac{2}{3}}}{L}V^5\right). \quad . \quad . \quad . \quad (7)$$

V = speed in knots per hour, D = displacement in tons, S = wetted surface in square feet, and L is the length between perpendiculars in feet; for f, n and b, see pages 17 and 18.

Coefficient of Propulsion.—The effective horse-power is that required to tow the ship. To find the power which must be developed by a steam-engine we must allow for the friction of the engine, the efficiency of the propeller, and for the interaction between hull and propeller. It is customary to lump all these in a single factor called the coefficient of propulsion, which varies from 0.45 to 0.65; that is to say the effective horse-power is only 0.45 to 0.65 of the indicated horse-power.

For turbine steamers and for internal combustion engines the shaft horse-power is reported and used in design. The coefficient of propulsion in this case is the ratio of the effective horse-power to the shaft horse-power. For turbine steamers the ratio is likely to be from 0.45 to 0.65, because the propellers chosen for such ships

have a poor efficiency. For ships and boats driven by internal combustion engines the ratio may run from 0.5 to 0.7; it does not appear to be so well known.

Mechanical Efficiency.—The mechanical efficiency of a steam-engine is the ratio of the power delivered to the propeller shaft to the power shown by the steam-engine indicator. This ratio depends on the workmanship and condition of the engine and shaft and may vary from 0.8 to 0.9. The larger value may be used for engines known to be in good condition.

Efficiency of Propeller.—The efficiency of propellers may be estimated from tables on pages 111 to 121, allowing for imperfections when necessary. For reciprocating engines under favorable conditions it may be taken as 0.65, for preliminary designs; for steam turbines it is liable to be as small as 0.50.

Hull-efficency.—The propeller from choice and necessity is placed at the stern of the ship where it works in the wake or stream of water set in motion by the ship. It can abstract some energy from the wake, a gain of five or ten per cent being possible from this action. On the other hand it disturbs the stream lines and the flow of water toward the propeller causes a reduction of pressure at the stern; it is popularly considered to produce a suction on the stern and thus to increase the resistance. This effect known as thrust deduction may amount to five or ten per cent. The wake gain and thrust deduction tend to counteract each other. To allow for this combined action it is customary to use a factor called hull efficiency which may vary from 0.9 to unity. For large well formed ships it is commonly taken as unity. A more complete statement of wake, thrust-deduction and hull-efficiency will be found on page 74.

The propulsion coefficient is the product of the mechanical efficiency, the propeller efficiency and the hull efficiency. If the mechanical efficiency is 0.9, the propeller efficiency 0.65 and the hull-efficiency is unity, then the coefficient of propulsion will be

$$0.65 \times 0.9 \times 1 = 0.6.$$

Problem.—Recurring to the problem first stated on page 6 we may compute the indicated horse-power for a ship to make 25

knots per hour by the independent estimate as follows. Basing the design on the *Campania* (page 8) we may first find the length from the corresponding speed

$$23.18 : 25 :: \sqrt{600} : \sqrt{L}, \quad \therefore L = 700 \text{ (nearly).}$$

The other dimensions as computed on page 7 should be beam 76.1 feet, and draught 29.2 feet. The displacement is found by the proportion

$$\overline{600}^3 : \overline{700}^3 :: 18000 : D, \quad \therefore D = 28600 \text{ tons.}$$

The wetted surface may be computed by the proportion

$$\overline{600}^2 : \overline{700}^2 :: 49620 : S, \quad \therefore S = 67540 \text{ sq. ft.}$$

The independent estimate is more flexible than either the Admiralty coefficient, or the theory of similitude, but is most successful when related to a type ship. In particular the factor *b* for the residual resistance should properly be deduced from speed trials of the type ship; but unfortunately it is not often determined or quoted.

If the main dimensions are determined in some other way or if they are modified from those derived from a type, then the displacement may be computed from the block-coefficient, and the wetted surface may be computed from equation (3), page 11. The block-coefficient should be the same or nearly the same as that for the type ship, and the length should vary but little from that determined by the law of corresponding speed. To complete the computation and exhibit the forms just quoted we may find this displacement and wetted surface as follows:

$$\text{Displacement} = 0.644 \times 700 \times 76.1 \times 29.2 \div 35 = 28600 \text{ tons.}$$

The ratio of beam to draught is

$$76.1 \div 29.2 = 2.6,$$

for which the factor C (page 11) is 15.51, so that

$$\text{Wetted surface} = 15.51\sqrt{28600 \times 700} = 69400$$

which is somewhat in excess of two per cent more than the wetted surface from the type ship by the theory of similitude; the former value (67,540) will be used in our computation.

The factor f and exponent n may be taken from Denny's table on page 17, as

$$f = 0.00847 \qquad n = 1.825.$$

Equation (7) applied to this case gives

$$\begin{aligned} \text{E.H.P.} &= 0.00307\left(0.00847 \times 67540 \times \overline{25}^{2.825} + 0.35\frac{\overline{28600}^{\frac{2}{3}}}{700}\overline{25}^{5}\right) \\ &= 0.00307 \times 0.00847 \times 67540 \times 8895 \\ &\qquad + 0.00307 \times 0.35 \times 935 \times 9766000 \div 700 \\ &= 15600 + 14000 = 29600. \end{aligned}$$

The computation is best made by aid of the tables of powers of displacements and speeds on pages 123 and 125. As a matter of convenience in the solution of the next problem the friction power and the residual power are computed separately and then added together.

The coefficient of propulsion may be assumed to be 0.6 and the indicated power may be estimated as

$$\text{I.H.P.} = 29600 \div 0.6 = 49300.$$

Model Experiments.—The fourth method for determining power is by aid of model experiments in a towing basin. To illustrate the method suppose that the tow-rope resistance for a paraffine model 20 feet long is 12.8 pounds, when towed at the corresponding speed of 4.23 knots. This speed is computed by the proportion

$$\sqrt{700} : \sqrt{20} :: 25 : V_m, \quad \therefore V_m = 4.23 \text{ knots}.$$

The theory of similitude gives for the wetted surface of the model

$$\overline{700}^2 : \overline{20}^2 :: 67540 : S_m, \quad \therefore S_m = 55.1 \text{ sq.ft.}$$

The friction factor and the exponent taken from Froude's table on page 17 are

$$f = 0.00834 \qquad n = 1.94,$$

consequently the frictional resistance is

$$0.00834 \times 55.1 \times 4.23^{1.94} = 0.00834 \times 55.1 \times 16.41 = 7.54 \text{ pounds.}$$

Subtracting this frictional resistance from the total tow-rope resistance of the model gives for the residual resistance

$$12.8 - 7.54 = 5.26 \text{ pounds.}$$

The corresponding residual resistance for the ship will be proportional to the displacement and the displacements are proportional to the cubes of the length, so that

$$\overline{20}^3 : \overline{700}^3 :: 5.26 : R_w, \quad \therefore \quad R_w = 225500 \text{ pounds.}$$

At twenty-five knots per hour the horse-power to overcome this resistance will be

$$0.00307 \times 225500 \times 25 = 17310.$$

This residual power added to the frictional power previously computed on page 25 will give for the total power

$$\text{E.H.P.} = 15600 + 17310 = 3290,$$

and with the coefficient of propulsion 0.6 the indicated power will be

$$\text{I.H.P.} = 32910 \div 0.6 = 54900.$$

Comparison of Methods.—The four several methods of estimating power given on pages 2, 12 and 24 may be compared as follows:

Admiralty coefficient	52,900
Law of comparison	53,300
Independent estimate	49,300
Model experiment	54,900

In this particular application the Admiralty coefficient and the law of comparison should give satisfactory results, because the type ship is supposed to be followed closely in the design. In passing from a smaller to a larger ship the tendency is to overestimate the power but not to a troublesome degree.

The independent estimate should be made to depend on trials of ships, the value of b being derived from an analysis of such trials, and may then be used with confidence. In the particular application, the factor b is probably underestimated because the speed-length-ratio is high for the *Campania*.

Under the most favorable conditions the determination of power from experiments on a model is liable to give a discrepancy from the power actually found on the ship after trial. Fortunately the discrepancy, which may be as large as ten per cent, is liable to be on the safe side. Designers who have sufficient information can usually estimate and allow for the discrepancy.

Methods for Small Boats.—Any of the methods of estimating power may be applied to small boats when there is sufficient information. Used with discretion the Admiralty coefficient will probably be found most direct and convenient. Some designers have been very successful in working up from smaller to larger boats by the theory of similitude. The experimental model should yield good results provided good sized models can be towed at sufficiently high speeds; in this case models less than the standard 10 or 20 feet may be used.

Fortunately an exact estimate of power is often of less importance for a small boat than for a ship, and a failure to realize speed is of less financial importance.

In some cases the owner or prospective purchaser will do well to invert the usual procedure, and having selected such a hull and engine, as he finds proper, may try to estimate the speed to be expected.

Keith's Method.—The following method of estimating the speed of a boat is due to Mr. H. H. W. Keith, instructor at the Massachusetts Institute of Technology; it has the peculiar merit that it uses only such dimensions as are commonly known for all boats and does not involve the displacement. The speed is computed by the equation

$$V=C\frac{\sqrt[3]{L\times P}}{B};$$

L is length over all in feet;
B is the extreme beam in feet;
P is the brake horse-power of the engine or engines.
The coefficient *C* is to be selected from the following table:

Type of Boat.	Ratio, $L \div B$.	*C*	
		Miles.	Knots.
Cruiser	3 to 5	9 to 11	8 to 9.5
Run about	5 to 7	8 to 10	7 to 8.5
High speed	.....	8 to 9	7 to 8

If the constant is taken from the column headed *miles*, then the speed is given by the equation in miles per hour; if from the column headed *knots*, then in knots per hour.

Problem.—Required the speed which will be given by a 10 horse-power engine to a cruiser having a length of 32 feet and a beam of $8\frac{1}{2}$ feet.

The ratio of length to beam is

$$32 \div 8.5 = 3.8.$$

This comes about half way between 3 and 5, so we may take the value of *C* half-way between 8 and 9.5 in the column for knots per hour, that is, $C=8.7$. The equation gives

$$V=8.7\frac{\sqrt[3]{32\times 10}}{8.5}=7 \text{ knots per hour.}$$

Had the constant been taken from the column for miles, its value would have been 10 and the speed would be 8.0 miles per hour.

As will be shown later, the equation conforms to the law of similitude and may therefore be used with confidence for similar boats at corresponding speeds provided that C is computed from a type boat; considerable divergence from type, and speed, will have comparatively little effect on the constant.

Example.—A boat 27 feet long over all and with 4 feet beam, which makes 14.5 miles per hour with 10 horse-power will have

$$C=\frac{4\times 14.5}{\sqrt[3]{27\times 10}}=9.$$

Wave Interference.—Attention has been called to the speed-length-ratio as a criterion of the relative speed of a ship or a boat, and it was said that in a general way ships and boats having a speed-length-ratio less than unity were relatively slow, while fast craft have a speed-length-ratio greater than unity.

In order to show how this division between fast and slow craft comes about and to emphasize the difficulty of determining power for high speeds a brief discussion will be given of the system of waves which travels along with the ship and the phenomena of wave interference.

A ship at high speed is accompanied by a system of waves which move with the same speed as the ship so that an individual feature, such as a particular wave crest, keeps the same position relatively to the ship. The most characteristic feature is the diagonal bow wave followed by a series of similar waves gradually spreading out in width and decreasing in height. These diagonal waves run away from the ship and have no part in wave interference.

Along the side of the ship and in the wake are a series of transverse waves of which the diagonal waves are the terminators. These transverse waves are approximately trochoidal in form and the length measured from crest to crest corresponds with the length computed by the theory of trochoidal waves for the speed of the ship.

In order to bring out clearly the relation of lengths and speeds of trochoidal waves and their relation to propulsion the following table has been computed:

SPEEDS OF TROCHOIDAL WAVES.

Length of Wave, Feet.	Square Roots of Lengths.	Time, Seconds.	Speeds.	
			Feet per Second.	Knots.
10	3.16	1.40	7.15	4.2
20	4.47	1.98	10.1	6.0
40	6.32	2.80	14.3	8.5
60	7.74	3.42	17.5	10.4
80	8.94	3.95	20.2	12.0
100	10.00	4.42	22.6	13.4
150	12.25	5.41	27.7	16.4
200	14.14	6.25	32.0	19.0
300	17.32	7.66	39.2	23.2
400	20.00	8.84	45.3	26.8
500	22.36	9.89	50.6	30.0
600	24.50	10.83	55.4	32.8
700	26.46	11.69	59.9	35.5
800	28.28	12.51	64.0	37.9
900	30.00	13.26	67.9	40.2
1000	31.62	13.98	71.5	42.3

The first column gives the length of the wave in feet and the second gives the square roots of the lengths. The third column gives the time required for a wave to run its own length. From the lengths and times, the speeds of the waves may readily be computed either in feet per second or in knots per hour. For example, the waves which accompany a ship at a speed of 19 knots per hour, are 200 feet long from crest to crest.

Thus far attention has been given to the bow waves only, but a similar system is formed at the stern, consisting of transverse waves with diagonal terminators. At a low speed the bow system and the stern system are practically separate, because the bow system is so spread out and diminished in height by the time it gets to the stern that it has then little effect.

When the speed of the ship increases so that the speed-length-ratio approaches unity, the bow waves preserve a considerable height at the stern and into the wake, where they combine with the waves of the stern system and wave interference becomes an important feature in the resistance of the ship. The nature of this phenomenon is most clearly seen from a study of its worst condition when the first well formed transverse bow wave crest comes in coincidence with the first stern-crest.

The bow wave at high speeds is irregular in form and is likely to be broken so that the location of its crest cannot be well

determined; it appears to be somewhat less than a quarter of a wave-length from the stem of the ship. The first stern wave is formed about a quarter of a wave-length from the stern-post; it is difficult to locate because it is not well developed at slow speeds, and at high speeds it is affected by the bow system. The distance from the bow wave to the stern wave is something more than the length of the ship between perpendiculars; it is estimated to be 1.05 to 1.15 of the length of the ship, and this is called the *wave-making length.*

The first well developed transverse bow wave crest is a wave-length from the bow crest. When the speed of the ship is such that the length of the trochoidal wave corresponding to that speed is equal to the wave-making length, then the bow wave, and stern wave coincide, resulting in the formation of a very high transverse wave in the wake of the ship.

Suppose that a torpedo-boat 182 feet long is running at 19 knots per hour; its wave-making length may be assumed to be 1.10, so that the length of the accompanying waves will be

$$182 \times 1.10 = 200 \text{ feet.}$$

The first well formed transverse bow-wave crest will coincide with the stern-wave crest and the boat will be in the worst condition for efficiency of propulsion. Up to a speed-length-ratio of unity, which in this case gives a speed of

$$\sqrt{182} = 13.5 \text{ knots,}$$

the power increases nearly as the cube of the speed; above that speed the power increases faster than would be indicated by the rule of cubes, and when the boat gets to 16 knots (half way from that speed to the worst speed) the power is likely to increase as the fourth power of the speed.

If the boat is driven faster than the worst speed the bow-wave crest draws astern of the stern-wave crest and the combination of the wave systems gives a more favorable condition. The most favorable condition would occur when the bow crest reaches the first hollow of the stern system, for then it would tend to suppress the formation of waves in the wake. Complete extinction of waves cannot however be expected.

In order to find the most favorable speed, we may note that the bow-wave and stern-wave systems will be half a wave-length apart and that they are separated by the wave-making length of the ship; that is to say the length of the waves will be twice the wave-making length of the ship. In the case chosen for illustration, twice 200 gives 400, for which the speed of the waves is 26.8 knots per hour. This boat may perhaps make 30 knots, which is well above the most favorable speed.

The conditions are laid down in an explicit manner because the phenomena thus appear to be simple; in reality they are not so simple and things cannot be expected to happen just as computed. But a complex system of phenomena may be comprehended better after a partial and simple statement has been made.

Power for High Speeds.—In any case the determination of the power required for propelling a ship at high speed is difficult and uncertain. Of the several methods of estimating power the theory of similitude is probably the best as the form for high speed must follow acknowledged good models. The problem is usually to get a higher speed with a larger boat, and will be solved by making the length proportional to the square of the speed; the power is then made proportional to the seven-sixths power of the displacement as explained on page 12.

It is desirable that a type ship shall be tried at various speeds from about half speed to full speed, the power for each speed being determined. This forms a progressive speed trial. By the aid of the theory of similitude the probable results of the progressive speed trials may be predicted in advance and represented by curves, and then as the trials progress the results may be computed and compared with the predicted results. Unusual and unfavorable features may be detected immediately and trials may be repeated or discontinued. Skilled builders find that results may usually be predicted with certainty.

Model basin experiments are very useful especially when new forms or conditions are to be provided for, especially in avoiding unfavorable combinations.

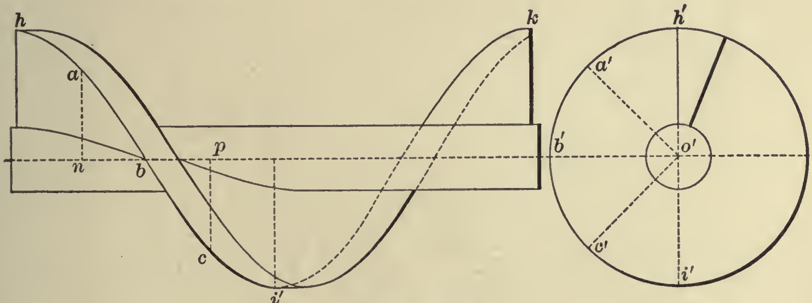
The Admiralty coefficient involves the theory of similitude and may be used with confidence for corresponding speeds and is a

fair guide for speeds that do not differ widely from that speed. Up to a speed-length-ratio of unity the coefficient changes but slowly with the speed. But as the worst speed is approached the Admiralty coefficient is to be used with caution. Well above the worst speed and to the most favorable speed the coefficient changes slowly and may be used to advantage.

The independent estimate may be used up to and somewhat above a speed-length-ratio unity, but not for high speeds.

Screw Propellers.—The only kind of propelling agent which will be considered in this book is the helical or screw propeller.

A true screw or helical surface is generated by a line which moves forward uniformly and revolves uniformly with one point in contact with a line called the axis.



FIGS. 3 and 4.

Fig. 3 represents one turn of a screw with a thin thread; the end projection is shown by Fig. 4. A quarter turn of the helix is shown by *abc*, *a'b'c'* of Figs. 3 and 4; the same figures are isolated in Fig. 5.

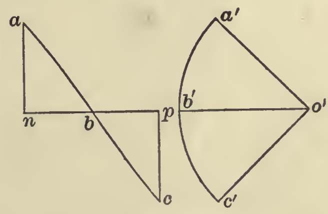


FIG. 5.

Sometimes the generatrix is inclined to the axis as shown by Fig. 6; the screw is then said to have a rake. The rake is usually aft to carry the blades of the propeller clear from the hull.

A helicoidal surface can be generated by a curved line like *oa* in Fig. 7. Special forms of screws with such peculiarities are made to conform to certain notions that sometimes are fanciful.

Pitch of a screw is the distance parallel to the axis between the successive threads. Variable pitches have been used for propellers and must be clearly understood.

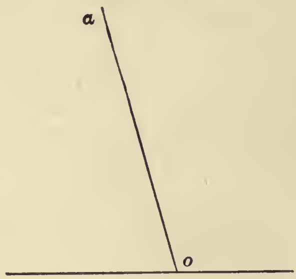


Fig. 6.

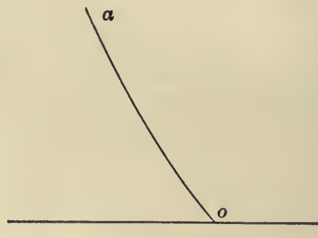


Fig. 7.

The pitch of a propeller blade may increase axially or radially. Fig. 8 shows a half-turn of a screw with axially increasing pitch. The generatrix revolves uniformly around the axis, but advances

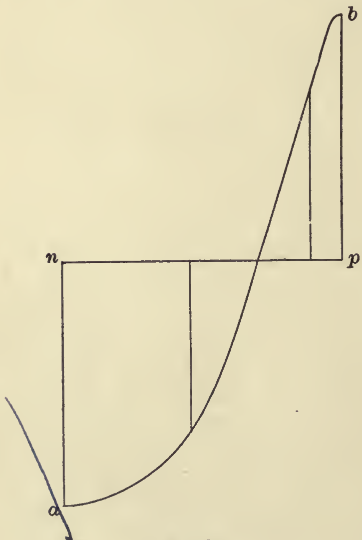


Fig. 8.

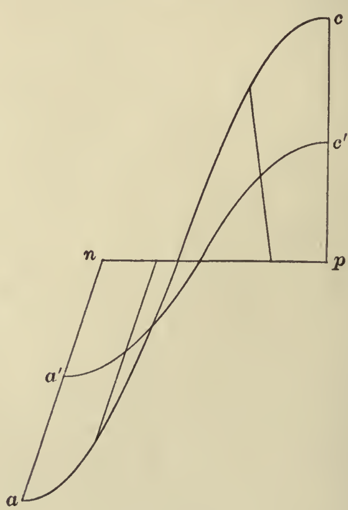


Fig. 9.

with increasing velocity from p towards n. A propeller with such a form is expected to accelerate the water gradually. There is some advantage from increasing axial pitch with very wide blades.

Fig. 9 shows half a turn of a screw with increasing radial pitch. The point *c* moves uniformly, generating a true helix, and the point *p* also moves uniformly but more slowly; the intermediate curves like *a′c′* are true helices. There does not appear to be any advantage from this device.

Projections of a Propeller.—Figs. 10 and 11 give projections of a four-bladed propeller with uniform pitch and no rake. It is represented as driving a ship toward the right. The left or after face of the propeller is a true screw, the blade thickness being put

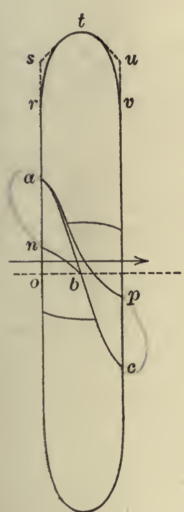


FIG. 10.

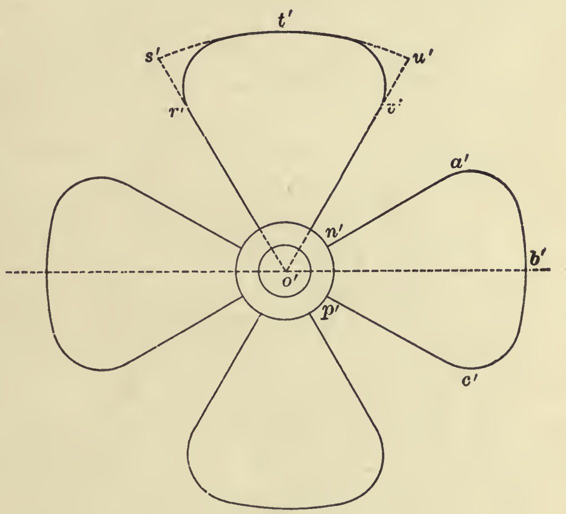


FIG. 11.

on the back. There is a practical advantage in making the face a true mathematical surface which can easily be constructed and verified. It is customary also to consider the pitch of the face of the propeller only, although the form of the back is as important.

The projection on a transverse plane, shown by Fig. 11, shows four blades each subtending 60°, that is, one-sixth of a turn of the screw. The contour *o′r′s′t′u′o′* shows the form which the blade would have if the helical surface were complete, with square corners; *orstu*, Fig. 10, is the projection of the same contour on a longitudinal plane.

To avoid vibration the corners are cut away, sometimes to a large extent; in Figs. 10 and 11 the corners are slightly rounded so that the helical forms shall not be obscured, and for the same reason the hub is a straight cylinder. The helical face intersects the hub in helices; the intersection of the back is more complex on account of thickness. In practice the hub is barrel-shaped or partly spherical. Propellers with straight edges and slightly rounded corners are but little used, because they have poor efficiency.

Proposed Standard Blade.—It will appear that the design of a propeller can conveniently be based on the projected area of the blade, as shown by Fig. 11, and for this purpose a standard projected contour is proposed, as it greatly simplifies the design. But the acceptance of this standard contour is not essential provided the contour selected does not vary in a marked manner from it.

Various theories of propellers have been based on a conventional development of the blades, and standard developed contours have been selected to go with the theories. But now that we have enough experimental information to avoid any other theory than that of mechanical similitude, we may save the labor of drawing the conventional developments by the simple expedient of choosing a standard projected contour.

The proposed standard blade contour is shown by Fig. 12. It has a cylindrical hub 0.2 of the diameter of the propeller, to correspond with the experimental propellers on which the propeller-tables are based, and also it provides a hub large enough for separable blades for three blades. The diameter of the hub may be increased if necessary or it may be made as small as convenient for a solid propeller, without much effect on the action of the propeller.

The remainder of the radius of the blade is to be taken as the major axis of an ellipse, which ellipse, together with tangents from the centre of the shaft, is to be taken as the projected contour. The major axis of the ellipse is therefore 0.4 of the diameter of the propeller. In Fig. 12 the projected contour is *vrtet'r'*.

Comparing this contour with that of a blade on Fig. 11, it is apparent that it differs in that the corners are very much cut

away, but the edges of the blade near the hub are elements of the helical surface. This conception is important because it is the basis of the method given later for drawing the projections of the propeller. The projected area of a blade is the area of the contour *vrtet'r'*, Fig. 12, in square feet. The area-ratio of a blade is the ratio of this projected area to the area of a circle having the diameter of the propeller.

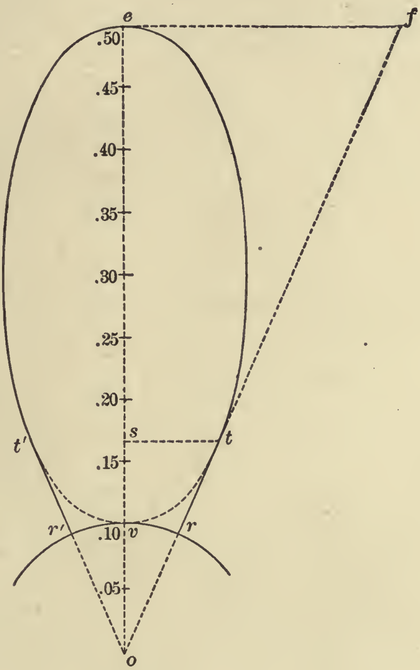


FIG. 12.

The projected width of a blade is measured at the minor axis of the ellipse, that is at 0.3 of the diameter of the propeller from the axis. The width may vary from about 0.2 to about 0.45 of the diameter of the propeller. When the width is 0.4 of the diameter the ellipse becomes a circle, as shown by Fig. 13; this circular contour is a convenient basis for determining properties of the propeller. If the width of the blade is more than 0.4 of the diameter, the width becomes the major diameter of the projected contour.

All the blades, of whatever width, that are obtained from the standard contour have that kind of similarity that comes from the choice of an ellipse for the contour. In particular the projected area of the blade is proportional to the width.

The straight edge of a blade, as *rt* Fig. 12, terminates at the point of tangency with the ellipse, that is, at *t*. To locate *t* make

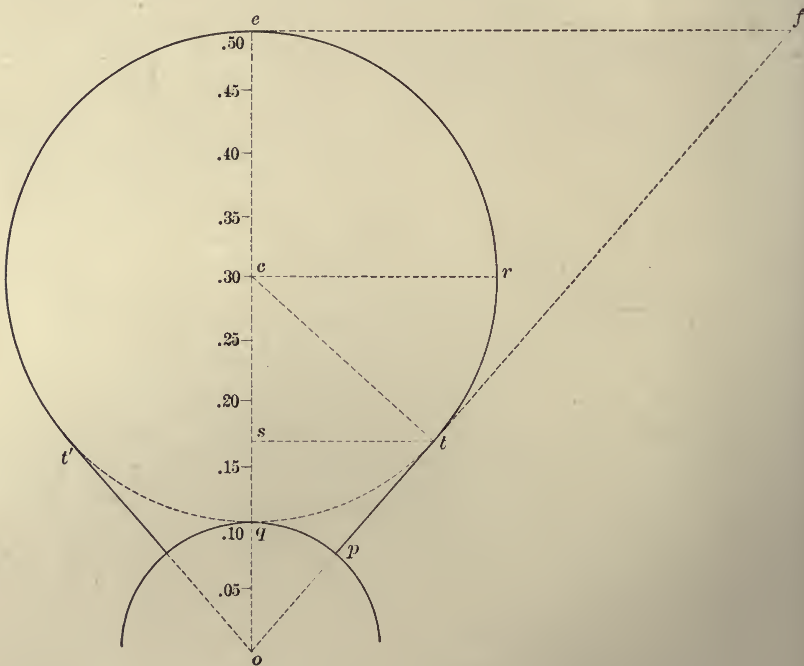


Fig. 13.

os equal to 0.1667 of the diameter and draw *st* perpendicular to *os* till it intersects the ellipse. The computation of this quantity and other convenient properties of the standard blade will be explained later. They may be accepted now without discussion; those who do not care to deal with minor theoretical points may accept them permanently.

It is convenient to lay off the angle *tos* by drawing a perpendicular at *e* to *oe* and laying off *ef*; the computation for this factor will be given later.

Laying-down Table.—If the standard projected blade contour is accepted, it is easy to compute and tabulate properties by aid of which propellers may be drawn with facility and precision.

LAYING-DOWN TABLE FOR ONE BLADE.

Area Ratio. One Blade.	Width Ratio.	Factor for Blade Angle.	Axial Factor.	Area Ratio. One Blade.	Width Ratio.	Factor for Blade Angle.	Axial Factor.
0.06	0.1456	0.1628	0.1002	0.14	0.3398	0.3799	0.2068
0.07	0.1699	0.1900	0.1156	0.15	0.3640	0.4070	0.2175
0.08	0.1942	0.2171	0.1304	0.16	0.3883	0.4341	0.2276
0.09	0.2184	0.2442	0.1447	0.17	0.4126	0.4613	0.2372
0.10	0.2427	0.2714	0.1583	0.18	0.4369	0.4884	0.2463
0.11	0.2670	0.2985	0.1714	0.19	0.4611	0.5156	0.2549
0.12	0.2912	0.3256	0.1838	0.20	0.4854	0.5427	0.2630
0.13	0.3155	0.3527	0.1956				

The laying-down table gives the following items: (1) the projected area-ratio for one blade; (2) the width ratio at 0.3 of the diameter; (3) the factor for laying down the blade-angle; (4) the axial factor.

The width-ratio multiplied by the diameter of the propeller gives the minor diameter of the ellipse in Fig. 12, which is laid off at 0.3 of the diameter from the axis.

The factor for the blade angle multiplied by the diameter of the propeller gives the line *ef*, which may be used for drawing the tangent *fo*.

The axial factor is to be multiplied by the pitch to find the axial width of the blade as shown in Fig. 10.

As already stated, the projected area-ratio is the ratio of the area of the contour *vrtet′r′*, Fig. 12, to the area of a circle having the diameter of the propeller.

Deduction of Properties.—The proportions and properties of the standard projected contour are deduced exactly by geometrical methods and may be accepted as correct, whether or not the reader cares to follow the deductions which are given here.

All the properties are readily computed for the circular contour shown by Fig. 13 and may be applied to any elliptical contour like that shown by Fig. 12 by the principles of projection. Suppose

that Fig. 13 were drawn on transparent tracing paper and held with the axis *oe* directly over and parallel to *oe* of Fig. 12; then suppose that Fig. 13 were turned to an angle about the axis *oe* till a given point in Fig. 13 (such as *f*) should come directly over the point indicated by the same letter in Fig. 12; then any other point (for example *t*) will also come over the point so lettered in Fig. 12, and the circle will lie directly over the ellipse. The circle is then said to be projected into the ellipse and in fact the whole of Fig. 13 is said to be projected into Fig. 12. Since all lines perpendicular to the axis, like *ef*, make the same angle they will be foreshortened to the same degree when projected onto Fig. 12. For example, if *ef* in Fig. 12 is half as long as the corresponding line in Fig. 13, so also will *st* be equal to half of the corresponding line. Since this relation holds for the half width of the ellipse taken at any distance from *o*, then the area of the ellipse in Fig. 12 is half that of the circle in Fig. 13.

The basis of comparison is the ratio of the semi-minor diameter of the ellipse of Fig. 12 to the semi-diameter of the circle of Fig. 13. Consequently having the properties of Fig. 13 we may get the corresponding properties of Fig. 12 or of any other elliptical contour, by multiplying by that ratio.

Tangent Point.—The straight edge of a blade ends at the tangent point of the ellipse of Fig. 12 or the circle of Fig. 13. The distance of the point *s* from *o* may readily be computed for Fig. 13 by drawing *ct* perpendicular to the tangent; then

$$oc : ct :: ct : cs.$$

$$\therefore\ cs = \frac{\overline{ct}^2}{oc} = \frac{(0.2)^2}{0.3} = 0.13333;$$

$$\therefore\ os = 0.3 - 0.13333 = 0.16667.$$

But since all the ellipses are obtained from the circle by projection, this relation holds for all the elliptical contours.

Projected Blade Area.—The half-blade of Fig. 13 can be divided into three parts: (1) the circular sector *ect*, (2) the triangle *oct*, and (3) the circular sector *oqp*, to be subtracted.

Begin by computing the angle cot from the equation,

$$\sin cot = \frac{ct}{oc} = \frac{0.2}{0.3} = 0.6667;$$

$$cot = 41^\circ\ 48'\ 37'' = 41^\circ.81.$$

The sector ect has the angle

$$ect = 90^\circ + rct = 90^\circ + cot = 131^\circ.81.$$

The area of a circle having the diameter 0.4 is 0.1257, and as the area of a sector is proportional to its angle,

$$\text{Area } ect = 0.1257 \times 131.8 \div 360 = 0.04601.$$

The triangle cot has the area

$$\begin{aligned} \tfrac{1}{2} oc \times st &= \tfrac{1}{2} oc \times ct \sin oct \\ &= \tfrac{1}{2} oc \times ct \sin (90^\circ - 41^\circ\ 49') \\ &= \tfrac{1}{2} \times 0.3 \times 0.2 \sin 48^\circ\ 11' = 0.02236. \end{aligned}$$

The area of a circle of the radius 0.1 is 0.031416, and the angle $qop = cot = 41^\circ.81$; consequently, the area of the sector qop is

$$0.031416 \times 41.81 \div 360 = 0.00365.$$

Adding the first two areas, and subtracting the third, and then multiplying by two for both sides of the blade gives

$$2(0.04601 + 0.02236 - 0.00365) = 0.1294,$$

which is the area sought for a circular blade; the area of a circle having the diameter unity is 0.7854; consequently, the area-ratio of the blade with circular projected contour is

$$0.1294 \div 0.7854 = 0.1648.$$

This is an important quantity for the standard blade, because all the properties of the blade are made to depend on it.

The projected area ratio for any projected width of blade is found by multiplying the ratio just computed, by the width-ratio and dividing by 0.4. Thus the width-ratio of Fig. 12 is 0.2; its projected area-ratio is

$$0.1648 \times 0.2 \div 0.4 = 0.08241.$$

Conversely, the width-ratio corresponding to any given area-ratio may be found by multiplying by 0.4 and dividing by 0.1648. Thus a blade having the area-ratio 0.08 will have the width-ratio

$$0.08 \times 0.4 \div 0.1648 = 0.1942.$$

The blade-area computed by this method is very nearly correct for propellers which have spherical hubs; if the hub is barrel shaped and the blade is narrow there may be an error of one per cent, a quantity which has no appreciable effect.

The total projected area-ratio for any propeller is found by multiplying the area-ratio for one blade by the number of blades.

Factor for Blade-angle.—In drawing the standard projected blade contour it is convenient to lay off the angle *eof*, Fig. 12, by aid of the dimension *ef*.

Turning to the circular blade contour of Fig. 13, we have for that case

$$ef = eo \tan cot = 0.5 \tan 41^\circ 48' = 0.44721.$$

For any other blade the factor may be made to depend on the width-ratio, or the projected area-ratio. By projection, the width-ratios and the dimensions *ef* are proportional. But the area-ratios are proportional to the width-ratios, so that the dimensions *ef* are proportional to the area-ratios. Thus the area-ratio 0.08 has the width-ratio 0.1942 as computed. The factor for *ef* is therefore

$$ef = 0.4472 \times 0.1942 \div 0.4 = 0.2171,$$

or

$$ef = 0.4472 \times 0.8 \div 0.1648 = 0.2171.$$

Axial Dimension.—Turning to Fig. 11 it will be remembered that the blades there subtend 60°, and have consequently one-

sixth of a turn of the screw; the axial width shown by Fig. 10 is therefore one-sixth of the pitch. In the same way the axial dimension of the blade in Fig. 12 will have the same ratio to the pitch that the angle *tot'* has to 360°. The laying-down table gives the dimension *ef*, and *ef* divided by *oe* gives the tangent of the angle *eof*; this is the half-angle and is to be divided by 180. There the factor for the blade-angle is 0.2171 for an area-ratio of 0.08, and the axial dimension factor is computed as follows:

$$0.2171 \div 0.5 = 0.4342 = \tan 23^\circ\ 28' = \tan 23^\circ.47;$$

$$23.47 \div 180 = 0.1304.$$

To Draw Projections.—Since all the dimensions and proportions can readily be computed for the standard projected contour, the designer will follow his judgment and habit whether he will make a drawing of the propeller or trust that to the makers. The following method will be found rapid and accurate. After the diameter and the projected area-ratio of the blade of a propeller have been determined by methods to be given later, the projections can be drawn as shown in Figs. 14, 15, and 16.

Let it be assumed that the propeller has four blades, a diameter of 10 feet, a pitch of 20 feet, and a projected area-ratio of 0.075 for one blade. By interpolation in the laying-down table the following dimensions can be found.

Width-ratio 0.1820; width $10 \times 0.1820 = 1.820$ ft. $= 21.84$ in.;
Axial factor 0.1230; axial dimension $20 \times 0.1230 = 2.46$ ft. $= 29.52$in.

In Fig. 14 the length is laid off equal to 5 ft., scale 1 in. = 1 ft.; and the radius of the hub is made $ow = 0.2 \times 10 \div 2 = 1$ ft. The line *we* is bisected at *h* and the width 21.84 in. is laid off from *x* to *y*. An ellipse is drawn with *we* and *xy* as the axes.

The dimension *ef* is computed as follows: after the factor 0.2036 is found in the laying-down table,

$$ef = 10 \times 0.2036 = 2.036 \text{ ft.} = 24.43 \text{ in.},$$

and is laid off on Fig. 14 and the line *of* is drawn; it is tangent to the ellipse at *t* and locates the straight-edge *ut* of the blade

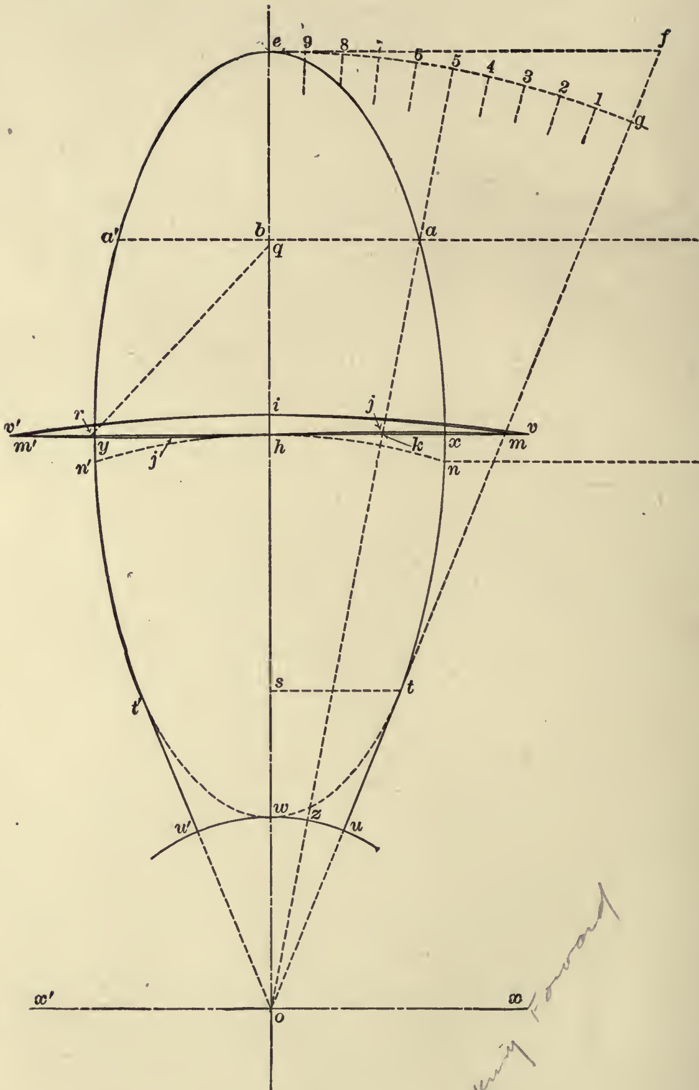


FIG. 14.

looking forward

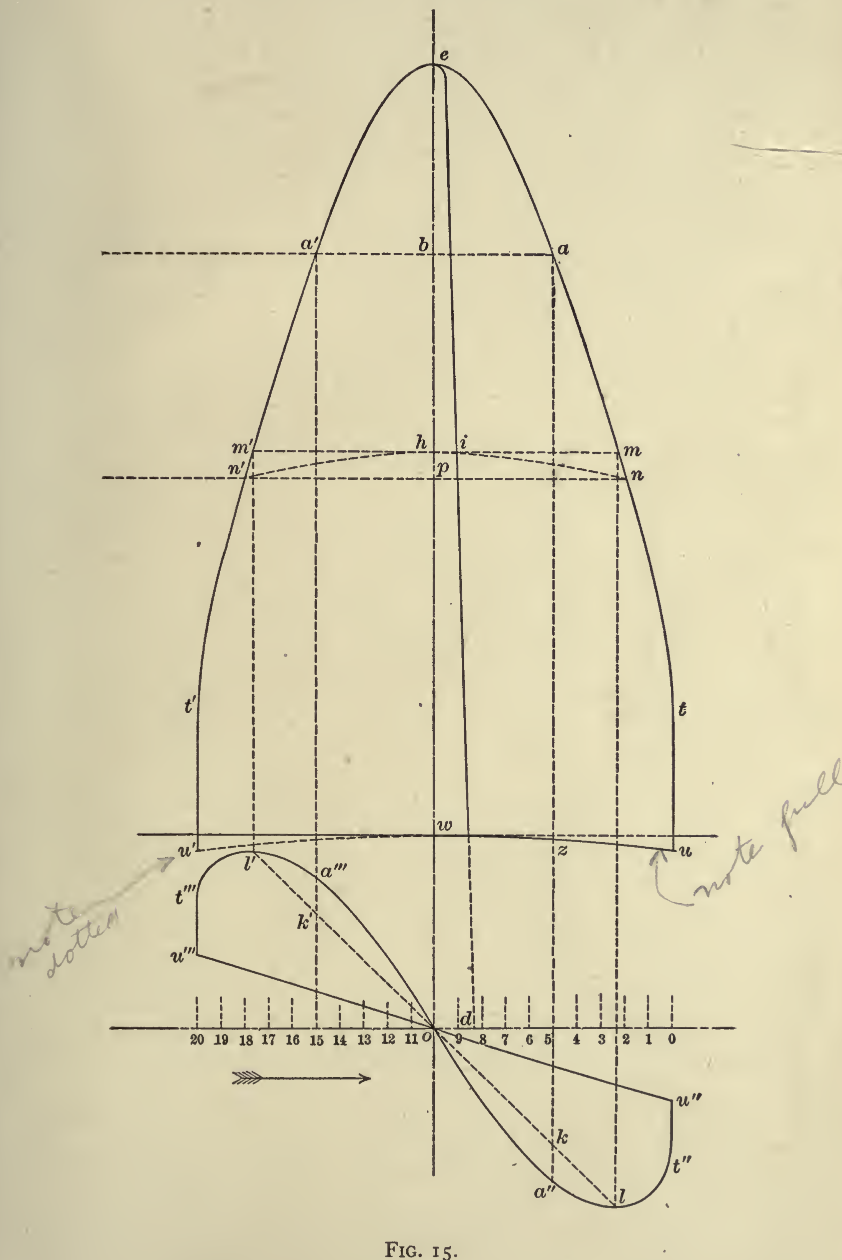

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FIG. 15.

contour. The line $u't'$ laid off on the other side of the blade completes the contour. In Fig. 14 the hub is drawn cylindrical, as shown by the arc uu'.

From the centre o the arc ge is drawn and divided accurately into ten parts by spacing with dividers; the arc on the other side of the blade gives a symmetrical construction and is therefore omitted.

The axial dimension uu' in Fig. 15 is laid off equal to 29.52 in. (scale 1 in. = 1 ft.) and is divided accurately into twenty equal parts and numbered consecutively from the right-hand or forward edge. This propeller, like that shown by Fig. 10, is right-handed and is represented as driving the ship toward the right. The left-hand surface or face is to be a true helical surface.

The point a on Fig. 14 is the intersection of the No. 5 radial line with the elliptical contour. On Fig. 15 this point is projected onto the No. 5 ordinate; the symmetrical point a' of Fig. 14 is projected onto the No. 15 ordinate. In like manner a sufficient number of points like a may be located and the contour may be drawn through them. It is now evident why the angle eog is laid off and divided with precision. The point t is accurately located by drawing st at 0.16667 of the diameter from the centre; in this case

$$os = 10 \times 0.16667 = 1.667 \text{ ft.} = 20 \text{ in.}$$

The straight line edges ut and $u't'$ of Fig. 14 appear as boundary elements ut and $u't'$ in Fig. 15. Since the hub is shown as a cylinder the root line of the blade is shown as a helical curve uwu'; a point like z is found by projecting z of Fig. 14 onto the corresponding ordinate; in the case shown the ordinate is No. 5.

The contour of a blade at right angles to that just described is shown by $u''a''oa'''u'''$ on Fig. 15. The point a'' is located on the fifth ordinate by making $5a''$ equal to ba of Fig. 14; the point a''' is symmetrical with a'' on the fifteenth ordinate. The bounding elements are $u''t''$ and $u'''t'''$ and the root line $u''ou'''$ is a part of a helix.

The drawing of the propeller for the information of the designer and the pattern-maker should be accurately drawn to a large

scale, if not full size. All lines should be drawn with a steel straight-edge; the axis of the ellipse and the ordinates should be laid off at right angles by a geometric method instead of depending on a triangle. The division of *eg* of Fig. 14 and of the axial dimension of Fig. 15 should be by spacing or some equally accurate method. A line through *e* parallel to the axis of the shaft should be laid off accurately and the divisions of the axial dimension should be transferred to it, so that the ordinates may be accurately located. The projection of a point, like *a* from Fig. 14 to Fig. 15, should be made by measurement; thus 5*a* should be laid off equal to *ob*.

The thickness of the blade, which is all applied to the back of the blade, is indicated by the line *eid*. The thickness *od* at the axis divided by the diameter of the propeller is known as the thickness ratio. In this case it is made equal to 0.02 of the diameter, so that the thickness is 0.2 of a foot or 2.4 inches. The thickness at the tip is 0.005 of the diameter, which in this case is 0.6 of an inch. Bronze blades are commonly made thinner at the tip; the thickness at the hub is greater for narrower blades. Cast-iron blades are much thicker.

Intersection at Hub.—For simplicity the hub is represented to be cylindrical and its intersection by the face of the blade is a helix. The hub is always a surface of revolution so that the intersection by an element of the face can be located by aid of a plane through it and the axis, which plane is to be revolved into the plane of the paper. The actual construction may be left to the draughtsman who will work to a large scale. In practice the blade joins the hub with rounded fillets cut by the pattern-maker.

Plane Section.—To show the form of the back of the blade and for the instruction of the pattern-maker, it is customary to give a number of sections like those shown on Fig. 14, where *mjhj'm'i* is a plane section and *vkhv'i* is a developed cylindrical section, to be explained in the next section.

A plane section perpendicular to the line *oe*, Fig. 14, cuts the contour at *xhy* and in Fig. 15 at *mhm'*; the points *m* and *m'* are projected to *l* and *l'*, and show the section of the blade contour

$u''t''ot'''u'''$, cut by a similarly placed plane parallel to the plane of the paper and at the distance *ob* above it. The plane section *lkok'l'* is shown in its correct form; it will be found to be slightly curved. To construct a point like *k*, draw the element *ok* 5 of the helical surface on Fig. 14 and note where it cuts the line *xhy* at the point *k*; this gives the correct transverse location of this point. On Fig. 15 draw the corresponding element 5*k* and make 5*k* equal to *hk* of Fig. 14. The symmetrical point *k'* is located by making 15*k'* equal to 5*k*. Having a sufficient number of points like *k* and *k'* the section *lkok'l'* can be drawn and transferred to Fig. 14. The thickness of the blade is laid off equal to *hi* and the back is drawn as the arc of a circle.

For cast-iron blades the edge cannot be so thin as this construction gives; so some thickness is given at the edge and then the back is rounded to the arc of a circle.

Very commonly the curvature of the line *mjhj'm'* is ignored in drawing plane sections of a blade because it is slight. The curvature is, however, important and must be allowed for, when sections are made to be employed for sweeping up the mould of a propeller on the floor of the foundry.

Developed Cylindrical Section.—Suppose that a cylindrical surface is constructed by revolving the line *mhm'*, Fig 15, about the axis of the shaft; it will cut the surface of the blade in a helix shown by *nhn'* and by the arc *nhn'* of Fig. 14. If the cylinder is developed into a plane the helix becomes a straight line. The development of the cylinder can be made in Fig. 14 by laying off the line *hr* equal in length to the arc *hn'*. The fore-and-aft dimension of the helix *hn'* of Fig. 15 is *pn'*. If this be laid off at *hq*, Fig. 14, the diagonal *qr* will give the half-width of the developed helicoidal section. This dimension is laid off at *hv* and *hv'*, and the back is drawn through *v*, *v'*, and *i*; for this purpose an arc of a circle may be used, though this is not quite correct if the plane section is constructed with the back rounded to the arc of a circle.

Sections like those discussed in this and in the previous section are drawn at intervals for the instruction of the pattern-maker; the choice of section depends on how the pattern is made. The draughtsman should have a practical knowledge of the making

of propeller patterns; there should be a competent person charged with the responsibility for the correct making of patterns and for maintaining them in correct form.

Blades with a Rake.—Fig. 16 shows the projection of the propeller of Fig. 14, but with 15° rake aft. The ordinates are now drawn with that inclination; the radius is measured perpendicular to the axis. In order to locate the helicoidal elements the helix $e'ee''$ must be constructed and then the elements like $0,e'$ and $20,e''$ can be drawn. The points a and a' of Fig. 14 may now be projected onto the proper elements at a and a' on Fig. 16. The contour of the edge of the blade $u''t''a''a'''t'''u'''$ can be drawn by the usual method of projections from Fig. 14 and the contour $utaa't'u'$; then the point a'' can be located on the vertical line aa'' at a distance $b''a''$ below the axis, this distance being equal to ba of Fig. 14. The thickness is laid off at right angles to the line $10,e$.

The cylindrical section $vkhv'i$ of Fig. 14 will be constructed as before, and will differ only in that the dimension hi will be slightly larger, because it is measured on a line inclined to the axis $10,e$ of the blade.

As for the form of the plane section, it will depend on how it is taken. If the plane is parallel to the axis of the shaft, the section will differ very little from that shown in Fig. 14, and that construction can be accepted for pattern-making or for sweeping up blades in the foundry; the sections in the foundry must in such case be set vertical, the blade being inclined at the angle of the rake from the horizontal. But if the section is perpendicular to the element $10,e$ as shown by nhn' of Fig. 16, the form will be materially different; it can be drawn by the ordinary methods of descriptive geometry, but the construction is omitted to avoid prolixity.

Helicoidal Area.—The true or helicoidal area of the blade of a propeller can be determined by aid of developed cylindrical sections, such as that which gives the line vhv' of Fig. 14; a number of such lines can be constructed at intervals from w to e, and a contour or bounding line can be drawn; the area of that figure will be the true area of the face of the blade. When the design

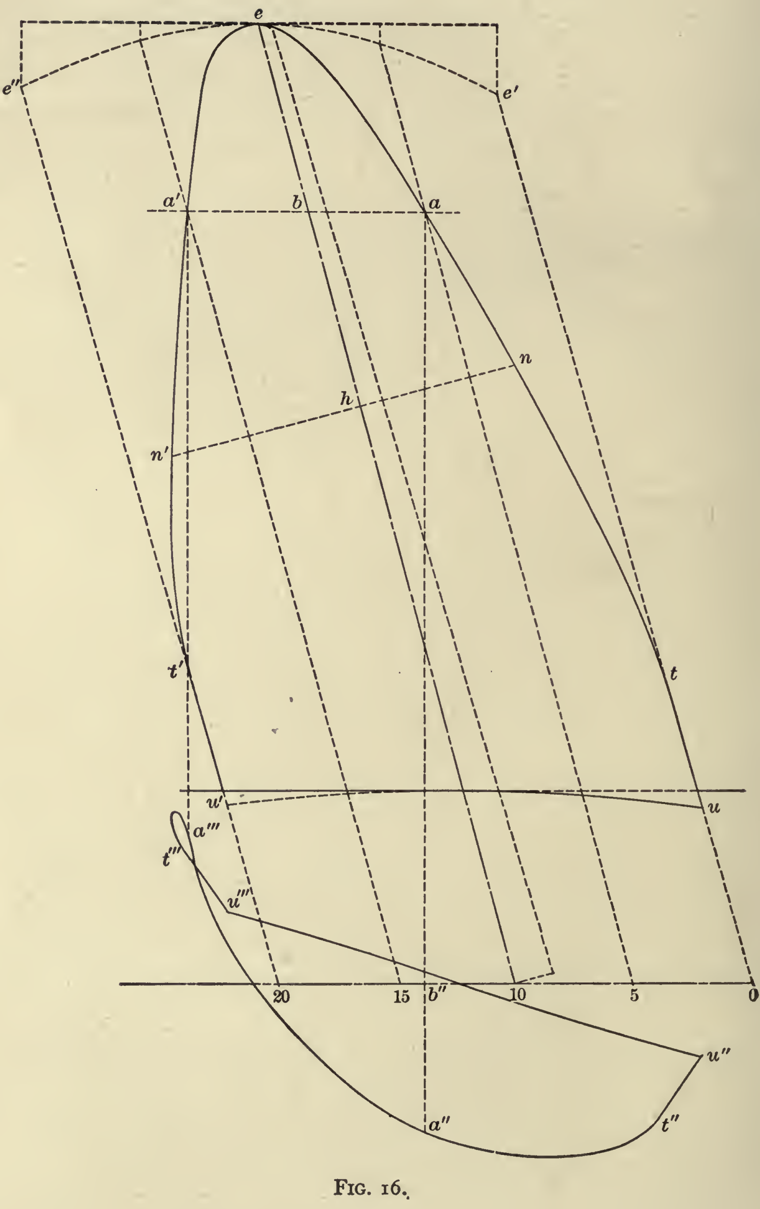

FIG. 16.

of a propeller is based on the projected area-ratio there is little reason for dealing with the area of the blade.

Developed Contour.—The surface of a screw-propeller is a ruled surface which cannot be developed, but there are conventional methods of constructing a plane figure which has nearly the same surface as a blade. These methods are called developing the blade, and the figure is called the developed contour.

The development of the blade of a propeller, and the inverse process of constructing the projections from the developed contour have an importance, because (1) certain propeller theories are based on the developed contour, (2) nearly all the experimental propellers tested in model basins have been designed from the developed contours, (3) and the results of such experiments systematized in tables and diagrams are stated in the same terms. In consequence engineers and designers are accustomed to working with the developed contour, and for that reason, if no other, the methods of drawing developed contour must be understood.

In Fig. 17 there is drawn half a turn of a helix *gabch* and the development *bf* of half a turn of the helix beginning at *b*. A quarter turn of the helical surface is shown by *nabcp*, comparable to the quarter turn shown on Figs. 3 and 4. The line *bf* is tangent to the helix at *b*; the deviation of the tangent at *s* from the helix at *c*, for an eighth of a turn is small; for less than an eighth the deviation is insignificant. Propeller blades seldom if ever are so wide as would be given by a quarter of a turn.

The conventional development of the blade of a propeller depends on the substitution of the straight line *bs* in place of the helical arc *bc*. The tangent *bs* is most conveniently located by drawing the triangle *tbu* in which *tu* is computed by the proportion

$$be : ef :: bt : tu.$$

But $be = \frac{1}{2}\pi d$, $bt = \frac{1}{2}d$, and $ef = \frac{1}{2}p$, where d is the diameter of the propeller and p is the pitch. Substituting and solving for tu,

$$tu = \frac{p}{2\pi}.$$

In Fig. 18 one-sixth turn of the helical surface is represented approximately by *nabcp*, in which *abc* is the tangent line in place of the true helical curve. Let a plane perpendicular to the plane of the paper be passed through the cylinder at *lbm*; it will cut an elliptical section of the cylinder which can be rotated into the plane of the paper, as shown on Fig. 19 by $ea''b'c''f$. The elliptical

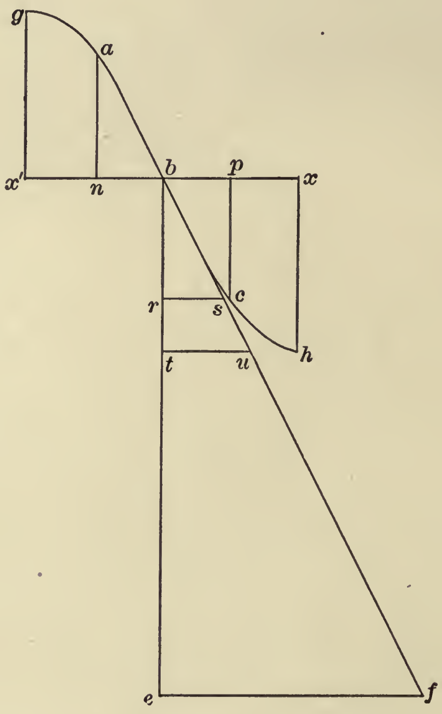

FIG. 17.

arc $a''b'c''$ is considered to be the development of the helical arc shown in projections by *abc*, Fig. 18, and $a'b'c'$, Fig. 19. Two other cylinders are represented by l_1m_1 and l_2m_2 in Fig. 18, with approximate helical surfaces a_1bc_1 and a_2bc_2; elliptical sections by planes through the line l_1m_1 and l_2m_2, are revolved into the plane of the paper in Fig. 19, thus locating the elliptical arcs $a_1''c_1''$ and $a_2''c_2''$. A curved contour is drawn through $a''a_1''$, $a_2''a_0$

and another through $c''c_1''c_2''c_0$. The points a_0 and c_0 are located by making

$$oa_0 = oc_0 = bp \text{ (of Fig. 18).}$$

The ellipses are all drawn from the foci o_1 and o_2, which may be located in the usual way; that is, by drawing arcs from b' with radii $b'o_1$ and $b'o_2$ each equal to oe. Or since the triangles

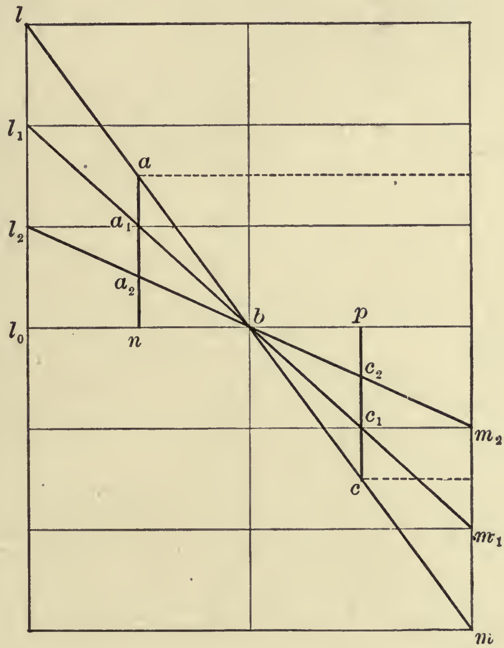

FIG. 18.

lbl_0 of Fig. 18 and $b'oo_1$ of Fig. 19 are equal to each other, the points o_1 and o_2 can be located by making

$$oo_1 = oo_2 = bl_0 \text{ (Fig. 18)} = \frac{p}{2\pi},$$

because bl_0 in Fig. 18 corresponds to tu of Fig. 17.

Another and simpler method is to take the lines ac, a_1c_1, and a_2c_2 of Fig. 18 and lay them off at kj, k_1j_1 and k_2j_2 on Fig. 20, and then draw the contour a_0kjc_0 for the developed contour of the blade. The contour $a_0a''b'c''c_0$ is repeated for comparison.

In designing propellers the developed contour is frequently drawn first and the projected contour is then constructed by reversing the methods just explained.

As an example we may refer to Fig. 25, page 58, which is given primarily to show the construction of a propeller with

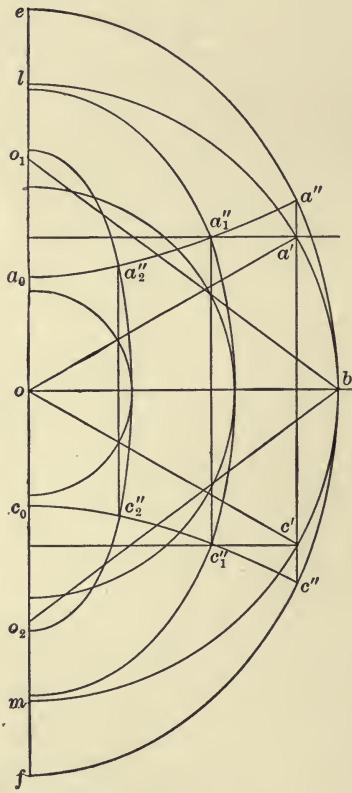

Fig. 19.

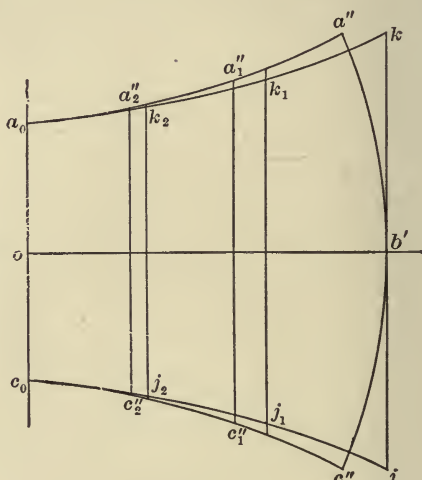

Fig. 20.

separable blades. The developed contour is shown by the dotted ellipse. *OA* is laid off equal to $p \div 2\pi$ to find the focus of the elliptical section, the point *A* corresponding to o_2 of Fig. 19. Choosing a point *B* we draw through it a circular arc *EB* from the centre *O* and an elliptical arc *DB*, with *OB* and *AB* for the semi-minor and semi-major axes. Through the intersection *D*

of the elliptical arc with the dotted contour, a horizontal line DF is drawn, which cuts the circular arc at E; this is a point of the projected contour. A comparison with Fig. 19 will justify this construction. A more precise method of locating points like F will be given in the description of Fig. 25.

Standard Developed Contour.—A form of developed contour for propeller blades which was first proposed by Wm. Froude and which is known as the Admiralty blade, is shown by Fig. 21. It

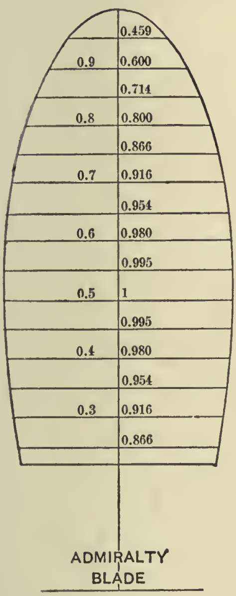

Fig. 21.

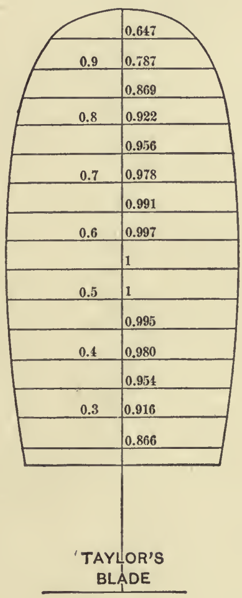

Fig. 22.

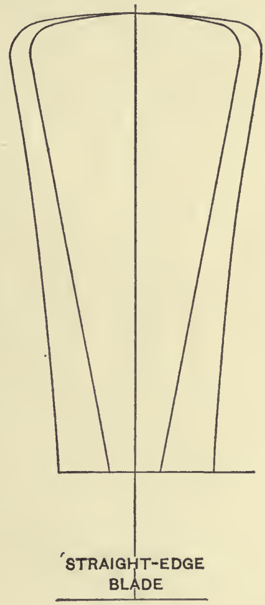

Fig. 23.

is an ellipse with the radius of the propeller as the major axis, and the minor axis is 0.2 of the propeller diameter. The diameter of the hub is 0.22 of the propeller diameter, and the contour is shown cut off by a straight line. More correctly the development of the root should be a curve depending on the form of the hub. With the advent of high-power and high-speed ships, especially turbine ships, the elliptical contour has been increased in width till it approaches a circle.

Fig. 22 shows a contour proposed by Naval Constructor D. W. Taylor, U.S.N., and used by him for many experimental propellers.

Its form is sufficiently determined by the ratios of the widths to the maximum width. Fig. 23 is put in to show the relative form of a straight-edged blade having about the same area, and slightly rounded corners.

In order to show the comparison of the proposed projected contour with the Admiralty blade, two developments by the conventional method are given by Fig. 24. The contour *ertml*

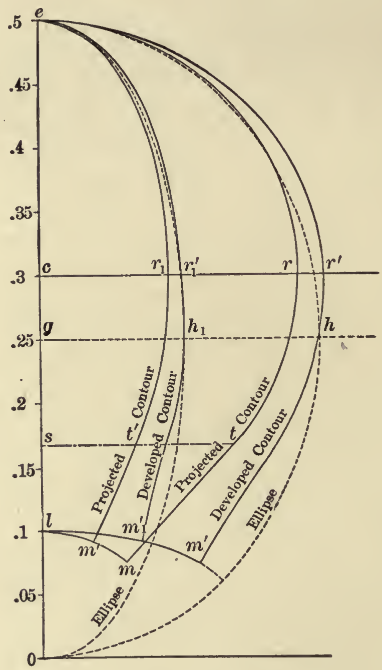

FIG. 24.

is the proposed circular projected contour, the radius *cr* being 0.2 of the propeller diameter so that the projected width is 0.4 of the diameter. The hub is 0.2 of the diameter and the contour at the hub is completed by a circular arc. The contour $er_1t'm'$ is drawn with the width equal to 0.2 of the diameter. The developed contours are drawn by the method of page 53, for a pitch-ratio of unity, that is, with the pitch equal to the diameter; a different pitch-ratio would have but little effect on the conclusion that

can be drawn from the figure. The dotted ellipses are drawn through the points h and h_1 on the line gh at the middle of the radius; they are the developed contours of the corresponding Admiralty blades. The developed contours shown by the full lines are wider at the tips and narrower at the hub; the area is somewhat less. Our design of propeller will be based on projected area-ratio which will set aside questions of width and area, but minor variations of either property have no appreciable influence.

Area of the Admiralty Blade.—The importance that is attached to the Admiralty blade makes it desirable to give ready means of determining both the developed and the projected areas.

The developed contour being an ellipse its area will be proportional to its width. If its width were half the diameter of the propeller, its area, neglecting the hub, would be 0.25 that of the disk or circle having the propeller diameter. The hub may be assumed to take away a segment having a rise 0.2 of the diameter of the circular contour; the segmental area is 0.1424 of that of the circle; consequently the net area is

$$0.25(1-0.1424)=0.214,$$

that of the disk. For any other width the area will be proportional; then for a width 0.2 of the diameter of the propeller the developed area is 0.0856 of the disk.

Barnaby gives the following rule for the projected area:

$$\text{Projected area}=\frac{\text{developed area}}{\sqrt{1+0.425\ (\text{pitch-ratio})}}.$$

This rule will give approximate results for other oval projected contours.

Construction Drawings.—The construction drawings for a four-bladed propeller with separable blades are shown by Figs. 25 to 28. The projection on a transverse plane looking forward is shown by Fig. 25, which gives also the developed contour on which the design is based. As previously explained OA is laid off equal to $p\div 2\pi$ and is the focus for the ellipses used in the development of the blade; or in the construction of the projected contour.

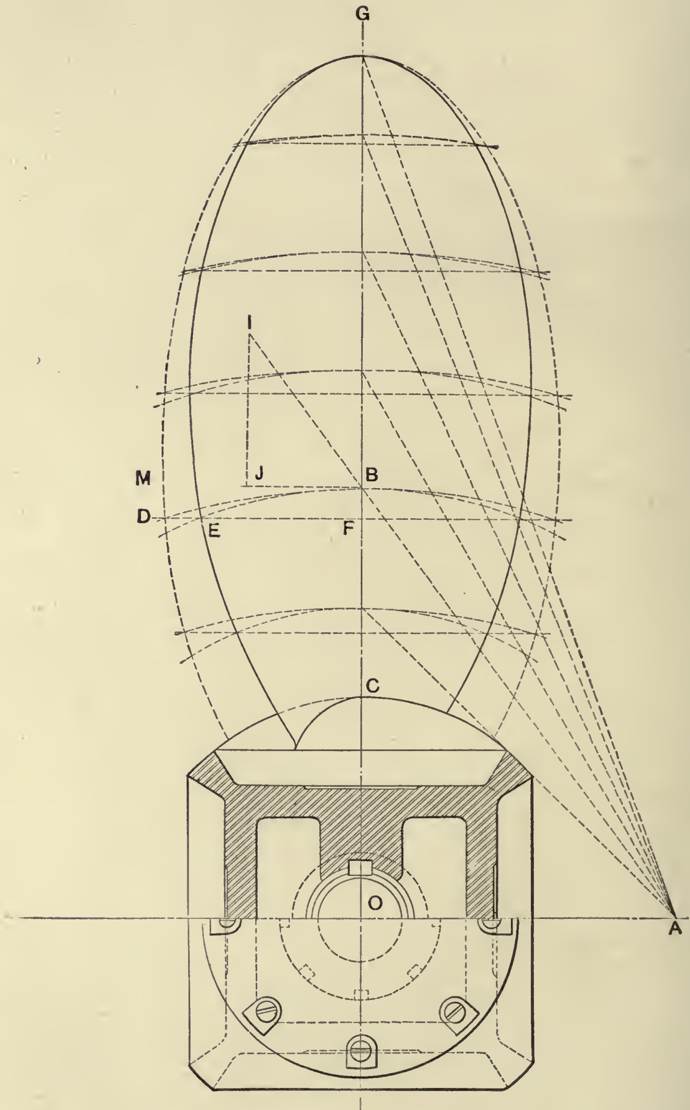

FIG. 25.

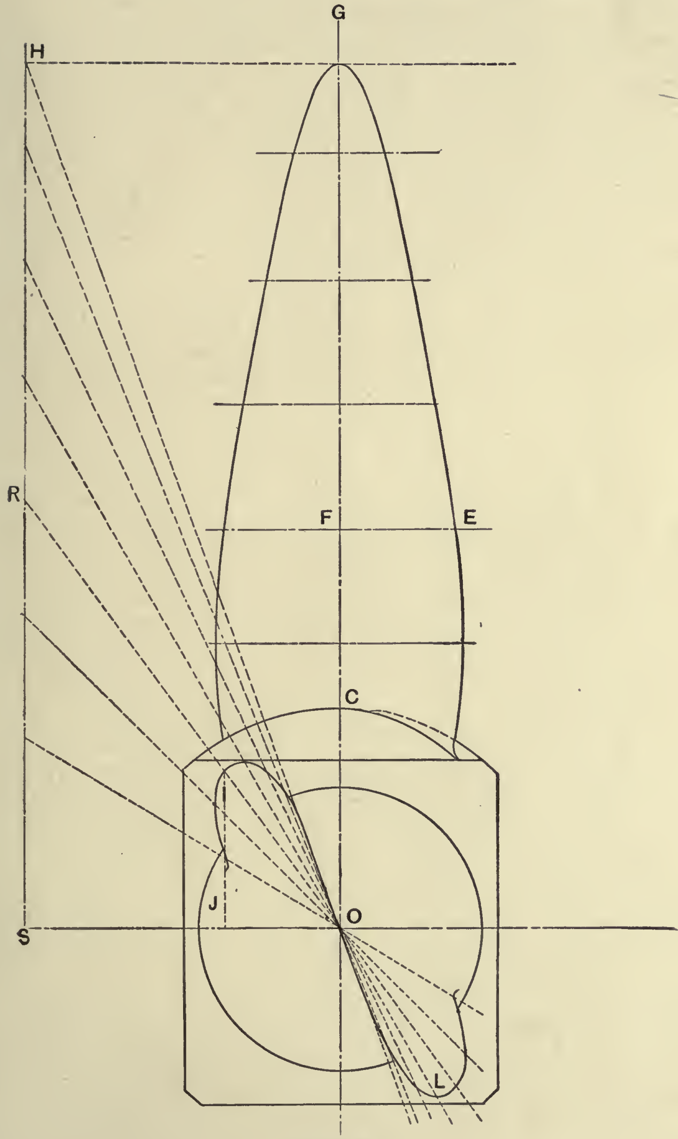

FIG. 26.

A point B is chosen and through it an elliptical arc is drawn; also a circular arc from the center O. A horizontal line DF locates the point E of the projected contour. Fig. 26 shows the projection of two blades, without rake, on a plane, through the axis of the shaft. OS is equal to OA of the preceding figure and SR is equal to OB; OI gives the projection of EF in its true length. Drawing IJ perpendicular to OS gives OJ, the proper half-breadth FE of the contour EG. It also gives IJ the proper length of EF of Fig. 25, and this is a more precise way of locating that point than that previously given.

The points I and J are points on the contour of a blade which presents its tip to the observer. The thickness of the blade and other details are omitted to avoid complexity.

Fig. 27 gives the projections of a blade with a rake, together with the effect of thickness of blade on the configuration of the tip and the root. The blades are fastened to the hub by flanges and bolts. Fig. 28 gives a longitudinal section of the blade and hub and shows details of construction.

Propeller Experiments.—The first systematic propeller experiments were made by the Froudes, father and son, at the Admiralty experimental tank, all being of the Admiralty type with the width of blade equal to 0.2 of the diameter. Mr. R. E. Froude has reported later experiments with various widths of blades.

Probably the most satisfactory tests are those by Naval Constructor D. W. Taylor, U.S.N., made at the model basin at Washington. The tables in this book are derived from these tests with the permission of Mr. Taylor. It has been shown that the tests by the Froudes and by Mr. Taylor are in substantial accord, so that both series of experiments may be claimed as the basis of the tables given in this book.

The tables for three-bladed propellers are based directly on an extensive set of experiments made on propellers of the Admiralty type with various widths, thicknesses, and pitch-ratios. The tables for four-bladed propellers were deduced from a comparison of tests on thin-bladed propellers of the type shown by Fig. 22 (some with three and some with four blades) with the tests on the Admiralty type. A table for two-bladed propellers

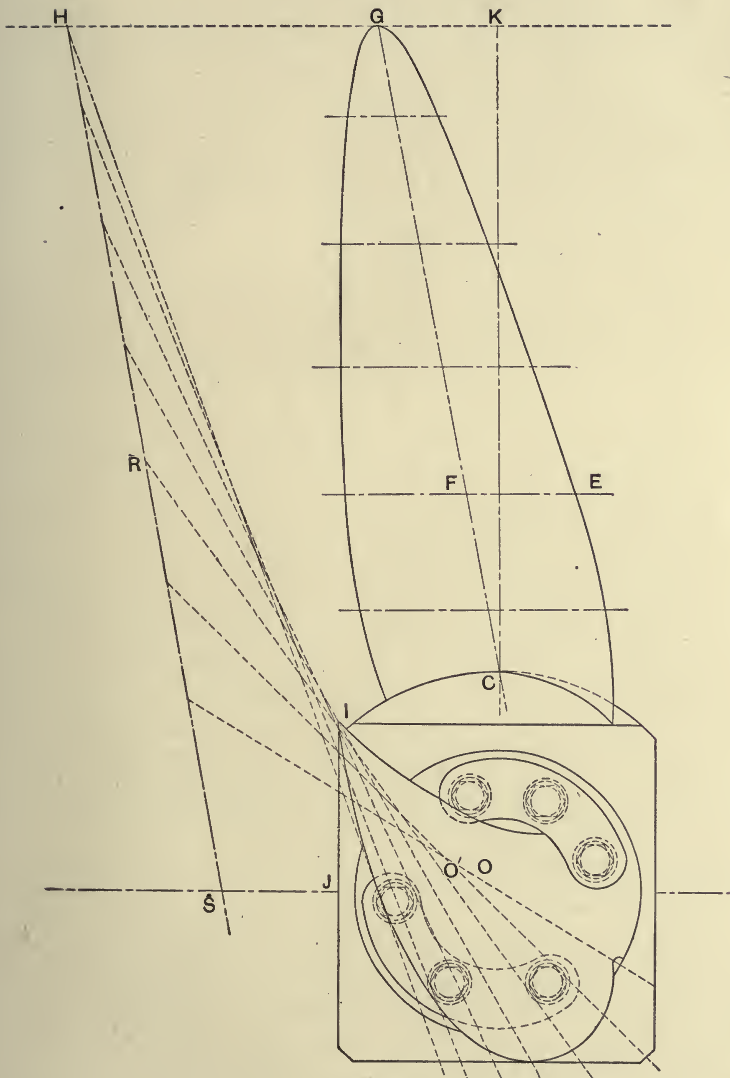

Fig. 27.

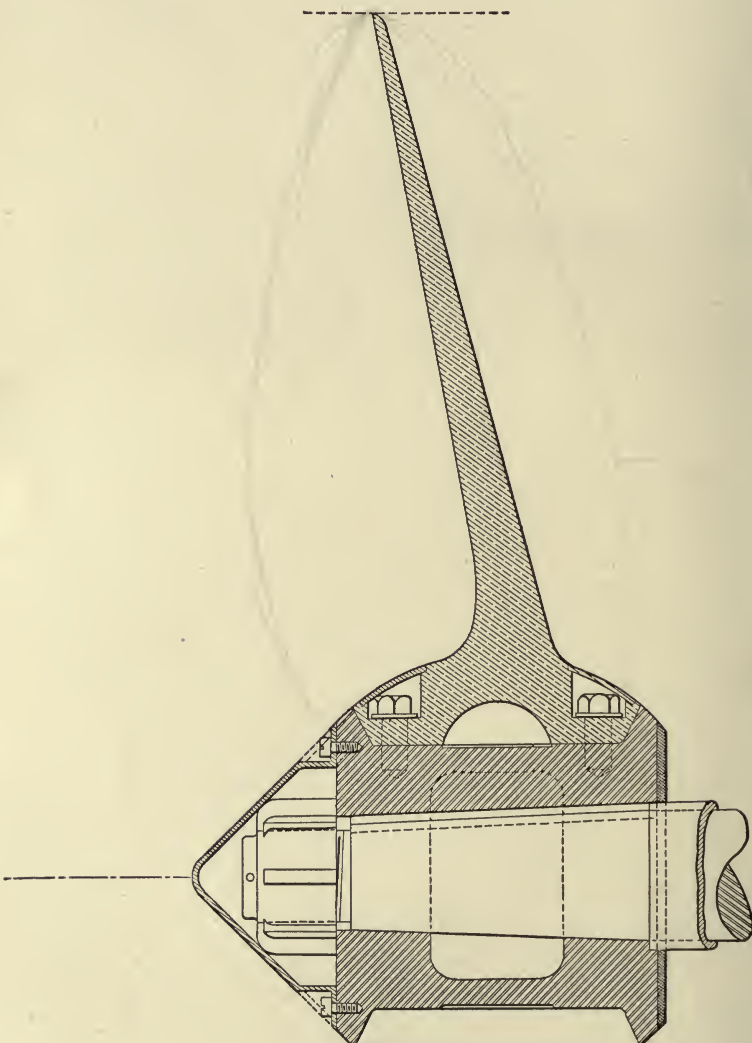

Fig. 28.

was deduced in like manner from tests of thin bladed propellers on that type.

Method of Experiments.—In making experiments in a model basin, the model propeller is placed at the front end of a shaft which is suspended from the towing carriage. The shaft at the rear extends into a boat-shaped box which contains the driving gear on the propeller shaft. The towing carriage is propelled at a convenient speed which is measured by appropriate devices. The propeller is driven at a convenient number of revolutions by some motor with arrangements for measuring the power required to drive it. The propeller pulls on the shaft and this force, which corresponds to the thrust of the ship's propeller, is measured; this force and the speed of the carriage give the data for the calculation of the power exerted by the propeller. To determine and allow for the friction of the driving gear and of the extruded part of the shaft, a test is made without a propeller on the shaft but with a filling piece shaped like the hub. After proper corrections and computations have been made the results can be stated in the form of the shaft horse-power required to drive the propeller and the propeller horse-power exerted by the propeller. The ratio of the propeller horse-power to the shaft horse-power is the efficiency of the propeller.

The method of determining the friction by a test without a propeller, but with a piece to replace the hub, has the effect of slightly underestimating the shaft horse-power, and consequently the efficiency is slightly overestimated; the effect is probably a small fraction of one per cent.

It is customary to make three or more runs with the same conditions; individual runs may vary as much as two or three per cent; the variations from the average is about half that amount. After a series of runs has been made with varying conditions, the results are represented by a fair curve. As two or more conditions may be subject to variation it is necessary to fair the results by the method of cross curves. The probable error of final results may be from half a per cent to one per cent.

Slip.—Let p be the pitch of a propeller in feet and let r be the revolutions per minute, then if it acted like a screw-gear working

in a fixed rack the speed would be pr feet per minute. Let the speed of the carriage be V_a knots per hour; then, since there are 6080 feet in a knot, the speed of the carriage is

$$\frac{6080}{60}V_a = 101.3\,V_a \text{ ft. per min.}$$

If this quantity is equal to pr it is considered that the screw-propeller does act as though it ran in a fixed rack. But in general the velocity of the carriage is less than pr, so that the relation is expressed by the equation

$$pr(1-s) = 101.3\,V_a; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

the quantity s is called the slip; it will hereafter be distinguished as the real slip.

Virtual Pitch.—The theory of internal propulsion indicates that a propeller can exert thrust and apply power only by imparting velocity to the water acted on. Now the slip is related to the action of imparting velocity and increases with that action. A natural inference would be that a propeller running without slip would exert no thrust, and this is nearly true for thin-bladed propellers which have the thickness equally distributed between the face and the back of the blade. If, however, the pitch used in calculating the real slip is that of the true helical face of the blade, then such a propeller will show an appreciable, and sometimes a large thrust with zero slip. Now the real action of the propeller blade on the water is an extremely complicated hydrodynamic problem, so that even qualitative conclusions must be drawn with caution. However, we may gain some insight into the matter under consideration if we consider that the action of a thick blade is comparable to that of a very thin blade having the form of the medial line, as shown in Fig. 29. Such a blade would have increasing axial pitch and the final acceleration would appear to be controlled by the pitch at the after edge. Since both width and thickness vary from tip to hub we cannot well assign a pitch on this consideration, but we can readily see why there is thrust at zero slip when the pitch is that of the

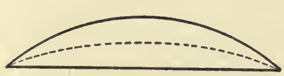

FIG. 29.

face. It has been proposed to assign to a propeller a *virtual pitch* which should be computed on the assumption that the slip is zero at zero thrust, by equation (8). It does not appear to be practical to base the design of propellers on virtual pitch, but the conception allows us to dispose of certain anomalies.

The question of virtual pitch and virtual slip is occasionally important; for example, it is desirable that the bow screw of a double-ended ferry-boat shall run idle and this can be accomplished by providing that there shall be no virtual slip. This condition is likely to obtain if the back of the blade is rounded because it becomes the driving surface for the bow screw.

Variable Pitch.—If it be considered that a propeller blade produces thrust by imparting acceleration to the water, it appears desirable that the blade shall have increasing axial pitch; this conception has exerted great influence especially on thoughtful engineers.

Now it is shown by experiments that there is a reduction of pressure ahead of the propeller and an increase aft of the propeller, the whole disturbance extending over a distance three or four times the diameter. The axial dimension of a propeller is small compared with this region of disturbance and the acceleration of the water while in contact with the propeller is only a fraction of the whole acceleration.

A propeller blade with a true helical face and rounded back may be considered to have increasing axial pitch; if the blade is narrow and thick the increase is excessive, and for this and other reasons the efficiency decreases with the thickness. There appears to be a slight advantage in dividing the thickness between the face and back of a propeller blade which has medium width. On the other hand wide blades with true helical faces show better efficiency with increasing thickness. Such blades if thin will have some advantage from increasing axial pitch. Mr. S. W. Barnaby says that very thin and wide blades may be crumpled at the forward edge when the thrust per square inch is high. Such blades may be designed with uniform pitch of the face at and near the after edge and then the pitch may be slightly decreased

toward the forward edge; there is no good guide for such a distribution of pitch.

Pitch-ratio.—The ratio of the pitch of a propeller to the diameter is called the pitch-ratio. It is one of the determining features of the design of a propeller.

Twisted Blades.—Large propellers are commonly made with separable blades, as shown by Fig. 25, page 58. They have the advantage that the pitch can be changed by twisting the blades. For this purpose the bolt holes in the flanges are elongated; filling pieces are provided so that the blade may be held securely. The development of the helix of Fig. 17, page 234, shows that the angle *ebf* is given by the equation,

$$\tan A = p \div \pi d_c,$$

where p is the pitch of the helix and d_c is the diameter of the helix. If the pitch is increased to p' the angle is increased, as shown by the equation,

$$\tan A' = p' \div \pi d_c.$$

By aid of this equation the following table was computed. The diameter of the flange of a blade (Fig. 28, page 62) in inches is to be multiplied by the factor given in the table, to find the distance measured along the circumference of the flange, through which the blade must be twisted in order to increase the pitch ten per cent.

Factors for Twisted Blades.—To increase the mean pitch ten per cent:

Pitch-ratio.	Factor.	Pitch-ratio.	Factor.	Pitch-ratio.	Factor.
0.6	0.0142	0.9	0.0191	1.4	0.0232
0.65	0.0151	1.0	0.0202	1.6	0.0236
0.70	0.0160	1.1	0.0212	1.8	0.0239
0.75	0.0169	1.2	0.0220	2.0	0.0243
0.80	0.0176	1.3	0.0227	....	....

For example, suppose the pitch-ratio is 1.2 and that it is desired to increase it ten per cent to 1.32, then the factor being 0.0220, a flange which is 40 inches in diameter should have a distance

$$40 \times 0.0220 = 0.880 \text{ inch},$$

marked off on its edge; and if the flange is turned through that distance the mean pitch will be increased ten per cent.

If the desired increase of pitch is less than ten per cent the distance marked off on the edge of the flange can be proportionally diminished. Thus, in the preceding example, the distance may be made 0.440 of an inch to increase the pitch five per cent.

If the distance is marked off backwards the pitch will be diminished nearly ten per cent, or a proportionally smaller amount for a less distance.

It is not advisable to increase or decrease the pitch more than ten per cent by this method, as it is approximate only and liable to decrease the efficiency.

The table has been constructed to alter the mean pitch ten per cent; the mean pitch being assumed to be that of the middle of the length of the blade, that is, at 0.3 of the diameter from the axis.

The construction of the table can be shown by computing one of the factors; for example, that at pitch-ratio 1.2. The diameter of the cylinder on which the helix at half-blade length lies is 0.6 of the diameter of the propeller,

$$\therefore \quad d_c = 0.6d.$$

The equation on page 66 gives

$$\tan A = p \div \pi \times 0.6d = 1.2 \div 0.6\pi = 0.6367$$

for the angle at pitch-ratio 1.2, while at pitch-ratio 1.32 the tangent becomes

$$\tan A' = 1.1p \div \pi \times 0.6d = 1.32 \div 0.6\pi = 0.7001.$$

The angles are therefore

$$A = 32^\circ\ 29'; \qquad A' = 35^\circ\ 0'$$

and

$$A' - A = 2^\circ\ 31' = 151'.$$

Now a circle one inch in diameter has a circumference of 3.1416, and 151′ will subtend an arc of

$$151 \times 3.1416 \div 60 \times 360 = 0.0220$$

of an inch.

Since the angle of the helix is smaller near the tip of the blade than near the hub, an increase of pitch by twisting the blade has relatively larger effect near the tip; consequently twisting a blade to increase the pitch gives the face an increasing radial pitch. On the other hand, the application of thickness to the back only, gives radially decreasing virtual pitch. One tendency counteracting the other, there is little harm in twisting the blade to increase the pitch. On the contrary, it is undesirable to decrease pitch by twisting the blade, a thing to be borne in mind in designing and adjusting blades.

Rake of the Blade.—The blades of a propeller are commonly raked aft to give them clearance from the hull. They may be raked aft as much as 15° without materially affecting the power or efficiency of the propeller. Raking the blades forward reduces the efficiency; fortunately there is no occasion for it. A raked propeller blade is longer than one without rake, and if it be made as thick it will weigh more. The worst effect, however, comes from the bending moment due to the eccentricity of the centrifugal force acting on the blade; quick-running propellers, like those for turbine steamers, should have no rake.

Blade Contour.—The oval blade contour is superior in efficiency to the wide-tipped type; but considerable variation in the form of the oval is allowable. The difference between the Admiralty type and Taylor's blade is inappreciable. The standard projected contour proposed falls within the limits of these two types, as shown by the development of Fig. 24, and Taylor's experimental results can be applied to it directly.

Thickness-ratio.—In Fig. 15, page 45, the lines of the face and back are extended to the axis; the ratio of the dimension *od* to the diameter of the propeller is called the thickness-ratio. In general, the thickness-ratio should be kept as small as may be consistent with strength. In order to provide sufficient strength the thickness must be greater for narrow blades, and as thick narrow blades are inefficient, a good width of blade will usually be chosen. But small propellers are commonly strong enough, so that narrow thin blades of high efficiency may be used for speed launches.

Form of Back.—As already indicated, the back of the blade, as shown by a section parallel to the axis of the shaft, is commonly rounded to the arc of a circle. Sometimes the section is parabolic or sinusoidal to give a sharp edge. Or the greatest thickness may be nearer the after edge for the same purpose. On the other hand, cast blades sometimes have considerable thickness at the edge. Propellers that are likely to work in floating ice may have blunt edges. Thick edges are likely to lose five per cent in efficiency if not more.

In much the same way the tip of a cast blade is given considerable thickness, as shown by Fig. 15, page 45. The longitudinal section of the blade may then have a straight back, as shown in the same figure. Sometimes the straight line of the back is drawn from *e* to *d*, and then the blade near the tip has a uniform thickness to favor the casting; this gives a hollow line near the tip. There is reason to believe that the greatest stress due to bending is found about 0.2 of the diameter from the axis. If this be accepted the greatest thickness should be located there, and the thickness might then be made uniform to the hub.

Tests of Similitude.—In order to investigate the application of the laws of similitude to propellers Mr. Taylor tested propellers having diameters of 8, 12, 16, 20, and 24 inches. All had the shaft 16 inches below the water level; the largest size consequently had the tip immersed only 4 inches, and the surface was appreciably disturbed, while the usual size of experimental propellers (16 inches in diameter) had an immersion of 8 inches, and showed no surface disturbance.

In general, the larger propellers absorbed relatively less power and had less efficiency than the small ones. The differences are not large and may be charged in part to the varying immersion. Mr. Taylor is of the opinion that the tests are favorable to the assumption that propellers follow the laws of mechanical similitude. Now the experimental propellers had three pitch-ratios, 0.6, 1.0, and 1.5; those having the largest pitch-ratios showed but little variation, and those having the smallest had not much variation. But the propeller having the pitch-ratio unity showed an appreciable variation, which may possibly aid in explaining

certain discrepancies between full-sized propellers and their models. Those propellers showed a loss of efficiency, the efficiency decreasing regularly from the 8-inch to the 24-inch sizes, the total difference being from three to five per cent. The 24-inch propellers required two per cent more slip than the 8-inch propellers in order to absorb the corresponding power. There is evidence that in some cases full-sized propellers show both less efficiency and less power absorbed than would be inferred from model experiments by the law of similitude. A few tests on full-sized propellers that would bear on this question would be very valuable.

Interaction of Propeller and Ship.—Thus far the propeller has been considered to act on undisturbed water, as a model does when carried on a frame in the towing-tank. When a propeller is placed behind a ship it acts on water which is disturbed by the ship, and, on the other hand, it disturbs the natural flow of water which closes in after the ship. This leads to the consideration of the wake and what is known as thrust deduction.

The Wake.—A ship propelled by sails or towed in undisturbed water, sets in motion a stream in the same direction; this stream or wake may be attributed mainly to the friction of the water on the skin of the ship. But near the stern there are other actions that may make the water move in the same direction and influence the wake at that place, namely, the stream-line flow and the effect of the transverse wave; also in some cases the wake may be affected by eddies. We may therefore consider that the wake may be attributed to

(1) Surface friction;
(2) Stream lines;
(3) Transverse wave;
(4) Eddies.

The predominant element in forming the wake is the surface friction; this can be seen from the fact that for all except very fast boats, the power to overcome frictional resistance is more than half the net horse-power, often it is two-thirds or more. This frictional wake is more intense near the middle and near the surface, diminishing sidewise and downward.

The whole subject of stream-lines whether considered theoret-

ically or practically is difficult and illusive. But both considerations show clearly that the pressure is higher near the bow and near the stern; in consequence there is formed the bow-wave and the stern-wave, each of which is about a quarter of a wave-length abaft the generating cause, which cause is the excess of pressure just mentioned. Now, just as in flow of water through a pipe an increase of pressure at the same level is due to the slackening of velocity. The water near the stern (which flows past the ship as the ship is driven through it) flows at a less relative velocity than the average, and consequently moves along with the ship, and contributes to making the wake. This influence is sensible near the ship but at a distance of a quarter of the ship's length is probably insensible.

Mention has been made of the transverse waves of the bow-system and the dependence of their location on the speed of the ship. When the crest of a transverse wave comes directly over a propeller, the water affected by the wave has a forward motion that extends to a considerable depth, gradually dying out. To illustrate the possible effect of such a wave on the wake it may be stated that a wave 200 feet long and which has a speed of 19 knots per hour, will have a velocity at the crest of 1.5 knots per hour, provided that the height of the wave from hollow to crest is 5 feet. This height is only one-fortieth of the length and is not excessive for the conditions found in practice. The speed dies away with increase in depth; at a depth of 5 feet the speed is 1.28 knots, at 10 feet it is 1.10 knots, and at 20 feet it is 0.80 knot; a rough average gives six per cent for the wake due to the wave in question. A shallow draught boat might have more than five per cent wake due to a crest of the transverse wave.

Conversely if there is a hollow of a transverse wave over a propeller the wake may be decreased six per cent or more in the case described above. Reports of zero wake or even of a negative wake are given by reliable authorities when there is a hollow over the propeller.

A well-formed steel ship should have no appreciable eddies, and should therefore not be affected by eddying wake. But there will be some eddying abaft propeller struts, and there may

be considerable effect from eddies near the webs for spectacle-frames of twin-screw ships, if those webs are set at unfavorable angles. There is in this case a partial compensation in that the propellers appear to be able to extract some energy from the eddies. Nevertheless, it is better to avoid such conditions unless the designer has full information from model experiments or otherwise.

A wooden ship with a wide stern-post shows a large and unfavorable eddying effect on a propeller set close behind it. If the stern-post cannot be narrowed then the propeller should be set well clear of the stern-post and a fair-water should be fitted to avoid eddies.

All these elements, namely, friction, stream-lines, waves, and eddies, tend to give a varying velocity to the wake. The wake will have higher velocity near the surface and near the axis of the ship. Now a propeller imparts kinetic energy to the water which is proportional to the square of the velocity imparted; in dealing with the influence of wake on the propeller we should therefore consider the squares of the effective accelerations produced by the propeller. But as such a method is impossible for various reasons, the wake is treated as though it were a uniform stream, which is equivalent to using the square of the mean acceleration instead of the mean of the square. Consequently, the efficiency of a propeller in a varying wake is likely to appear to be higher than in the open water, and such an effect is reported by Froude, but as the effect is small he recommends that wake be treated as uniform.

The mean value attributed to the wake of a large well-formed ship by Froude is ten per cent of the speed of the ship. The wake factor is the ratio of the velocity of the wake to the velocity of the ship, and is represented by w. Froude's mean value for w is 0.1; this is to be used for twin-screw ships; single-screw ships are likely to have more wake.

There is very little known about the wakes of large ships either as to the velocity or its distribution. The values reported for wake have been derived from experiments in the towing-tank, first on propellers in the open water and then on the same pro-

pellers properly placed behind models; the computations will be explained later.

Real and Apparent Slip.—The slip of the propeller as defined on page 63 gives

$$s=\frac{pr-101.3V_a}{pr}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

where V_a is the speed of the carriage in knots per hour, p is the pitch in feet and r is the number of revolutions per minute.

The conditions for a propeller working in a uniform wake can be inferred from what would happen if the water in the tank could have a forward velocity imparted to it equal to the speed of the carriage multiplied by the wake factor. Suppose that the speed of the carriage is now V knots per hour and that the wake factor is w; the speed of the water would be wV knots per hour, and the speed of the propeller through the water will be

$$V_a \doteq V-wV=(1-w)V \quad . \quad . \quad . \quad . \quad . \quad (10)$$

knots per hour. This speed of the propeller through the water may be called the velocity of advance. So far as the propeller is concerned it will behave just as though it were driven through still water from a carriage with the speed V_a. For a given real slip computed as before by equation (9) it will require the same torque and will deliver the same thrust. The work delivered to the propeller will be the same because the torque and revolutions are unchanged; but the work delivered by the propeller will be larger because the thrust will now act through

$$101.3V=101.3V_a \div (1-w) \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

feet per minute.

Apparent Slip.—If a ship is driven at a speed of V knots per hour by a propeller having a pitch of p feet, and making r revolutions per minute, the apparent slip is the quantity computed by the equation

$$s_1=\frac{pr-101.3V}{pr}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

If the wake of the ship is assimilated to a uniform stream then a propeller astern of the ship may be assumed to have a speed of advance of

$$V_a = (1 - w)V,$$

and its properties may be inferred from those of a model propeller having the real slip computed from this speed of advance.

From equations (9) and (12) the relations of wake factor, real slip, and apparent slip can be determined, and expressed by the equation

$$1 - s = (1 - s_1)(1 - w). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

It is to be remembered that s_1 is the apparent slip computed from the speed of the ship, w is the wake factor, and s is the real slip which depends on the speed of advance of the propeller through the water.

Wake Gain.—It is evident that there is a material gain in placing the propeller astern, where it can get the advantage of the wake. This comes from the fact that the thrust on the thrust-block works at the speed of the ship; the thrust as previously explained depends on the speed of advance. The gain from working the propeller in the wake is

$$\frac{V}{V_a} = \frac{V}{V(1-w)} = \frac{1}{1-w}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

The wake gain is really due to the fact that the propeller is able to extract from the wake a small part of the power expended by the ship in making the wake. Though the advantage of working in the wake is properly utilized, a greater advantage comes from anything that will reduce the wake.

Thrust-deduction.—If the screw-propeller could be placed a considerable distance behind the ship, it might get the advantage of working in the wake without disturbing the stream-lines about the ship; but it is necessary for various reasons to place the propeller well under the stern; consequently, the propeller disturbs the stream-lines and reduces the pressure at the stern. This reduction of pressure is equivalent to an increase in resistance, so

that it takes more power to propel a ship than it would to tow it. It is customary to represent the increased power required to overcome this action by aid of a factor,

$$\frac{1}{1-t}. \quad \ldots \ldots \ldots \quad (15)$$

Hull-efficiency.—The ratio of the wake gain to the factor for thrust-deduction

$$\frac{1-t}{1-w}. \quad \ldots \ldots \ldots \quad (16)$$

is called the hull-efficiency. Now, while both wake and thrust-deduction may be appreciably different for a ship and its model, they vary in somewhat the same way, so that the hull-efficiency is likely to vary less than the elements from which it is derived. Moreover, the hull-efficiency for large well-formed ships will not be very different from unity, and unless we have direct evidence, we may therefore commonly leave it aside in powering ships and designing propellers.

Determination of Wake and Thrust-deduction.—There are two ways of investigating the factors for wake and thrust-deduction, namely, by model experiments in the towing-tank and by the analysis of progressive speed trials.

Model experiments in the towing-tank, as outlined, are made by these three operations, or their equivalents:

(1) The model is towed with all appendages in place, but without the propeller, to determine the resistance R at the speed V.

(2) The propeller is adjusted behind the model and is driven at such a number of revolutions r as will develop a thrust T equal to the pull of the model at the speed V; on account of the thrust-deduction the pull is now greater than R.

(3) The propeller is run in the open water at the same number of revolutions r, and the speed of the carriage V_a is adjusted so that the thrust shall be T as in the second operation.

The thrust-deduction is then found by the equation,

$$\frac{T}{R}=\frac{1}{1-t}; \qquad \therefore\ t=1-\frac{R}{T}.$$

The wake is

$$\frac{V-V_a}{V}=w.$$

Since it is difficult to secure the exact adjustments given above it is customary to make a series of experiments for each condition and to select the quantities derived from faired curves, the details are a matter for the experimenter to adjust and need not be considered at length here.

The operations for finding wake and thrust-deduction are purposely stated in the form which is convenient for calculation rather than for experiment, in order to clarify the conceptions of those properties and to emphasize the fact that they are the properties of models; the corresponding properties for ships may be inferred from those for models, but with considerable difficulty and uncertainty.

In the first place it is difficult to get sufficiently certain and exact information for ships even after careful and exhaustive trials; but when the trials are satisfactory so far as they go, they are necessarily incomplete. Thus, for reciprocating engines, it is necessary to allow for the friction of the engines, of which but little is known positively; for turbine steamers the shaft horsepower is found directly, and in so far there is less uncertainty. The feature in which trials are necessarily incomplete is the power delivered by the propeller to the thrust-block.

Even so explicit a matter as pitch of the propeller may be uncertain, either because the pitch may vary or because the measurement of the pitch may have been slighted. Planed propellers are of course free from this difficulty.

When we undertake to infer the wake and thrust-deduction for a ship from its model it is necessary to use the theory of similitude, which is known to fail for the resistance and may be suspected for the propeller. In particular it is known and allowance is made for the fact that surface friction does not follow the laws of similitude. In consequence the slip of a model propeller must be larger than the slip of the ship's propeller; the apparent slips are known to vary in this manner, and the real slips *may* vary more markedly.

From these considerations it is clear that in order to make towing-tank results of real value they must be a part of a system including trials of the ships after construction. From such a system certain factors can be determined by which it is possible to infer with sufficient certainty for practical purposes what a ship will do from tests on its model. Very commonly all the factors are lumped into one called the coefficient of propulsion, defined on page 22.

A statement of methods of making progressive speed trials the observations to be taken, the precautions to be observed, and the deductions from them will be found in the author's *Naval Architecture*. Fortunately, a reasonably good approximation to the wake of the ship is sufficient for the design of the propeller.

Factors for Wake and Thrust-deduction. The factors which are given for wake and thrust-deduction are mainly those reported from time to time by R. E. Froude, which were deduced mainly for war-ships, some of which are of obsolete types. Recently an extensive series of experiments were reported by Mr. W. J. Luke for twin-screws applied to a common form of merchant ship.

Both Froude and Luke report that the number and area of the blades of a propeller have little effect on either wake or thrust-deduction. Luke reports that increased diameter increases both wake factor and thrust-deduction, but considers that the effect is rather due to changes in clearance between the propeller and and the hull than to the increased size.

The change of clearance between the propeller and the hull has a great effect on both wake and thrust-deduction; insufficient clearance is always to be avoided.

Pitch-ratio has an appreciable but not important effect on both factors.

Change of speed of the model had practically no effect on thrust-deduction, but the wake decreased appreciably with increasing speed. For a speed-length-ratio

$$\frac{V}{\sqrt{L}}=0.8,$$

which is common for such a type of ship; the wake was about 0.17, and the thrust-deduction was about 0.16, so that the hull efficiency was somewhat more than unity.

An approximate determination of the wake of a *model* may be made by the equations:

Single-screw ships

$$w=0.20+\tfrac{1}{2}(\text{block-coefficient}-0.55).$$

Twin-screw ships

$$w=0.10+\tfrac{1}{2}(\text{block-coefficient}-0.55).$$

The wake of a large ship is likely to be less than the amounts given by these equations, perhaps as much as ten per cent. An allowance of ten per cent would make the first term 0.10 instead of 0.20 for single-screws and would reduce that term to zero for twin-screws.

Mechanical Efficiency.—A marine engine may be expected to lose from 10 to 15 per cent of its power in friction, variously distributed at the pistons, crank-pins, main-bearings, thrust-block, and elsewhere; the power required to drive the air-pump from the main engine is variously estimated from 3 to 7 per cent. The mechanical efficiency may consequently be estimated from 0.8 to 0.9. Experiments with torsion meters from a few engines in good condition with independent air-pumps have shown efficiencies from 0.9 to 0.92; though there are difficulties in applying torsion meters to reciprocating engines, it is fair to assume that engines may have an efficiency of 0.9 under favorable conditions. There appears to be no reason why this factor should be affected by size, but rather that it depends on the construction and condition of the engine.

Effective Horse-power.—The simplest and perhaps the most useful information that can now be derived from a towing-tank is the resistance of the hull with appendages. Let the resistance of the ship as computed from model experiments, be represented by R in pounds. Then if the speed of the ship in knots per hour is V

the speed in feet per minute will be 101.3V; the effective horse-power will then be defined as

$$\text{E.H.P.} = R \times 101.3V \div 33000 = 0.00307RV. \quad . \quad . \quad (17)$$

If the resistance is estimated in some other way than by direct experiment on the model, the same form may be used to compute the effective horse-power.

Coefficient of Propulsion.—The coefficient of propulsion is taken as the ratio of the effective horse-power to the indicated horse-power,

$$\text{Coefficient of propulsion} = \text{E.H.P.} \div \text{I.H.P.}$$

For turbine steamers the shaft horse-power may be substituted for the indicated horse-power, bearing in mind that the mechanical efficiency does not enter into the coefficient.

The connection between the effective horse-power and the indicated horse-power can be built up in the following manner:

If e_m is the mechanical efficiency the power delivered to the shaft will be

$$\text{S.H.P.} = e_m \times \text{I.H.P.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

The shaft horse-power multiplied by the efficiency of the propeller e_p will give the power charged to the propeller. But the propeller gains from the wake, so that the power applied to the thrust-block is

$$e_p \times \text{S.H.P} \times \frac{1}{1-w}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

On the other hand, the interference of the propeller with the stream-lines increases the resistance and consequently the power required for propulsion is

$$\text{E.H.P.} \times \frac{1}{1-t}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

The expressions (19) and (20) must be the same, so that finally,

$$\text{Coefficient propulsion} = \frac{\text{E.H.P.}}{\text{I.H.P.}} = e_m e_p \frac{1-t}{1-w}, \quad . \quad . \quad (21)$$

that is, the coefficient of propulsion is the continued product of the mechanical efficiency, the efficiency of the propeller, and the hull-efficiency.

If the hull-efficiency is assumed to be unity and if the efficiency of the propeller is assumed to vary from 0.5 to 0.7, while the mechanical efficiency is taken from 0.8 to 0.9, the coefficient of propulsion may vary from

$$0.8 \times 0.5 = 0.4 \quad \text{to} \quad 0.9 \times 0.7 = 0.6.$$

The factor is commonly taken as 0.5 to 0.55 for well-formed ships; this should usually give a margin for contingencies.

Method of Reporting Experiments.—The Model Basin at Washington undertakes tests of models of propellers for private parties, under certain restrictions, and as the results are reported in a particular way, it is proper to present it here. Usually the information is in the form of curves plotted or real slips as abscissæ and gives the efficiencies at various slips, and also the factor A for computing the shaft horse-power by the following equation,

$$\text{S.H.P.} = A \frac{d^2 V_a^3}{1000}; \quad \ldots \ldots \quad (22)$$

where d is the diameter of the propeller in feet and V_a is the speed of advance in knots per hour, while A is a factor that varies with the slip.

A model to one-fifth natural size of the propeller of the U. S. Revenue Cutter *Manning* was tested at the Basin with the results given in the following table:

MODEL EXPERIMENTS ON "MANNING" PROPELLER.

Real slip	0.0	0.02	0.04	0.06	0.08	0.10	0.12	0.14
Value of A	1.74	1.96	2.20	2.48	2.80	3.13	3.48	3.86
Efficiency	0.587	0.615	0.640	0.654	0.665	0.673	0.678	0.683
Real slip	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30
Value of A	4.29	4.79	5.34	5.93	6.60	7.75	8.17	9.05
Efficiency	0.682	0.680	0.677	0.672	0.668	0.660	0.652	0.644

The *Manning* on trial had an apparent slip of 13.5 per cent at 16 knots per hour, and special experiments indicated that the

wake was 7 per cent. By equation (13), page 74, the real slip was

$$s=1-(1-0.135)(1-0.07)=0.20.$$

The above table gives at 0.20 real-slip $A=5.34$. The diameter of the propeller was 11 feet, and consequently equation (22) gives for the shaft horse-power,

$$\text{S.H.P.}=\frac{5.34\times 11^2\times 16^3(1-0.07)^3}{1000}=2130,$$

the speed of advance being

$$V_a=(1-w)V=(1-0.07)16$$

from equation (10) on page 73.

From the indicated horse-power on trial the shaft horse-power was estimated to be about eight per cent less than the amount computed as above. Discrepancies of this nature under the most favorable circumstances between computations from model experiments and data from trials, are not unusual. Reasons for the discrepancies can often be assigned and allowances can sometimes be made which will reduce or remove apparent discrepancies. But experienced designers who are familiar with model experiments usually prefer to let the discrepancy stand and to allow for it *en bloc* when they have occasion to predict trial results from experiments. There is good reason for taking the small wake factor 0.07 for the *Manning;* were it proper to take the more common value of 0.10, the discrepancy would appear to be disposed of.

The form of report of experiments on propeller models is convenient for comparison with trials of the ship, and its propeller; it is not convenient for the selection of a propeller for a particular service.

Propeller Tables.—The tables at the end of this book will be found convenient for determining the dimensions and proportions of propellers; they may ordinarily be used without interpolation.

To enter the tables first compute the revolution factor R by the equation,

$$R=\frac{r^{\frac{1}{2}}(\text{S.H.P.})^{\frac{1}{4}}}{V_a^{\frac{5}{4}}}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

r = revolutions of the engine per minute;
S.H.P. = the shaft horse-power, to be estimated from the indicated horse-power when necessary;
V_a = velocity of advance of the propeller to be estimated by the following equation,

$$V_a = (1 - w)V; \quad \ldots \ldots \quad (24)$$

V = speed of the ship in knots per hour;
w = wake factor.

Fortunately, a considerable variation of either power or wake factor will have relatively small effect.

Having computed R, enter any of the tables for two, three, or four-bladed propellers and find the value of the diameter factor D corresponding. Then compute the diameter by the equation,

$$d = \frac{D(\text{S.H.P.})^{\frac{1}{6}}(V_a)^{\frac{1}{6}}}{r^{\frac{2}{3}}}; \quad \ldots \ldots \quad (25)$$

D = tabular value corresponding to R of equation (23);
S.H.P. = shaft horse-power;
V_a = speed of advance of propeller;
r = revolutions of engine.

It is to be borne in mind that there are two places to be pointed off in tabular values of R and one place in D.

Problem.—Required the dimensions for a propeller for a ship which is driven at 16 knots by an engine which develops 3000 horse-power at 100 revolutions per minute.

Taking 0.9 for the mechanical efficiency gives for the shaft horse-power,

$$0.9 \times 3000 = 2700.$$

The speed of advance of the screw with a wake of 0.1 will be

$$V_a = V(1 - w) = 16(1 - 0.1) = 14.4.$$

The revolution factor will therefore be

$$R=\frac{(100)^{\frac{1}{2}}(2700)^{\frac{1}{4}}}{(14.4)^{\frac{5}{4}}}=\frac{10\times 7.21}{28.05}=2.57.$$

The four-bladed table, page 112, area-ratio 0.36 gives $D=51.4$ at 1.3 pitch-ratio and 0.2 slip. Consequently the diameter is

$$d=\frac{51.4(2700)^{\frac{1}{6}}(14.4)^{\frac{1}{6}}}{(100)^{\frac{2}{3}}}=\frac{51.4\times 3.73\times 1.56}{21.54}=13.9,$$

$$p=1.3\times 13.9=18.1 \text{ ft.}$$

The apparent slip is computed by the equation

$$1-s_1=(1-s)\div(1=w)=(1-0.2)\div(1-0.1)=0.889; \quad \therefore \quad s_1=0.11.$$

The powers required for solution of this problem are most readily obtained by interpolation in the tables on pages 122 and 123, after which the numerical computation can be made by aid of a slide rule.

If preferred the solution may be made by logarithms as follows:

$$\begin{array}{r}
\log\ 100=2.0000\\
\tfrac{1}{2}\\
\hline
1.0000\\
0.8578\\
\hline
1.8578\\
1.4480\\
\hline
\log 2.57=0.4098
\end{array}
\qquad
\begin{array}{r}
\log 2700=3.4314\\
\tfrac{1}{4}\\
\hline
0.8578
\end{array}
\qquad
\begin{array}{r}
\log 14.4=1.1584\\
5\\
\hline
4)\overline{5.7920}\\
\hline
1.4480
\end{array}$$

$$\begin{array}{r}
\log 2700=3.4314\\
\log 14.4=1.1584\\
\hline
6)\overline{4.5898}\\
\hline
0.7650\\
\log 51.4=1.7110\\
\hline
2.4760\\
1.3333\\
\hline
\log 13.9=1.1427
\end{array}
\qquad
\begin{array}{r}
\log\ 100=2.0000\\
2\\
\hline
3)\overline{4.0000}\\
\hline
1.3333
\end{array}$$

Problem.—Required the dimensions of twin-screw propellers for a ship to be driven at 20 knots by two engines each developing 8000 horse-power at 90 revolutions per minute. Here

$$\text{S.H.P.} = 8000 \times 0.9 = 7200,$$

$$V_a = 20 \times 0.9 = 18,$$

$$R = \frac{90^{\frac{1}{2}}(7200)^{\frac{1}{4}}}{18^{\frac{5}{4}}} = 2.36.$$

At pitch-ratio 1.5 and real slip 0.24 in the table for three-bladed propellers area-ratio 0.27, page 117, this corresponds to $D = 50.5$, and

$$d = \frac{50.5(7200 \times 18)^{\frac{1}{6}}}{90^{\frac{2}{3}}} = 17.9 \text{ feet},$$

$$p = 1.5 \times 17.9 = 26.85 \text{ feet},$$

$$s_1 = 1 - \frac{1 - 0.24}{1 - 0.1} = 0.156.$$

Choice of Conditions.—There is apparently a wide range of choice given the designer by the tables on page 111 et seq.; though conditions are limited in practice and sometimes narrowly, the designer usually has a considerable range which may at first seem confusing. There are, however, a number of conditions that can be stated simply to guide choice.

Number of Blades.—Large single-screw ships habitually have four-bladed propellers.

Ships with two, three or four screws usually have three-bladed propellers. Sometimes two propellers out of four have three blades and the other two have four.

Small craft of all sorts commonly have three-bladed propellers. Sometimes they have two-bladed propellers.

Area-ratio.—The projected area-ratio for one blade may commonly be taken as 0.09; three-bladed propellers then have a total area-ratio 0.27 and four-bladed propellers have 0.36.

If there is danger of cavitation (a term to be explained later) larger area-ratios are selected.

Narrow blades are useful mainly for small craft and may give comparatively high efficiency.

Best Efficiency.—The best efficiency is indicated in the table by printing values of R in full-faced type. Values of R as computed by equation (23), should be located as near such full-faced type as possible, but moderate deviations have little effect on efficiency.

Pitch-ratio and Slip.—In the presentation of this method of selecting a propeller for a given purpose, pitch-ratio and slip appear to enter incidentally or as matters of secondary importance. In reality they are of first importance and the experienced designer has a very good idea of the conditions desirable for his problem. Fortunately, the method here proposed will usually lead to customary relations.

For large ships the pitch-ratio will range from 1.0 to 1.5 and the apparent slip from 0.10 to 0.20; both pitch-ratio and slip increasing with the speed-length-ratio. Turbine steamers suffer from the necessity of using a high number of revolutions and a small pitch-ratio, the latter being commonly 0.7 to 0.8.

Efficiency.—The efficiency in the neighborhood of the full-faced type ranges from 0.45 to 0.75, increasing with the pitch-ratio, and being larger for narrow blades and for propellers with few blades (three or two). But the variation for a given type of propeller is not large and can be known approximately in advance.

Small Diameter.—The most common restriction on the design of a propeller is the necessity to use a small diameter with a ship of a given draught. This is the main reason for using four-bladed propellers for single-screw ships. For the same reason wide blades may be chosen, but they give little advantage except as a means of avoiding cavitation.

Having selected the number of blades and the area-ratio, special conditions, such as small diameter, can be sought by using other parts of the table remote from the full-faced type. Thus if we should on page 112 take a pitch-ratio of 1.50 and a slip of

0.26, the value of D becomes 48.3 and the propeller diameter will be 13.1 instead of 13.9 as computed in the problem on page 83; the efficiency is now 0.67 instead of 0.68.

If these several devices fail to give a propeller small enough for the conditions, then the revolutions of the propeller must be increased and the problem stated anew.

Precautions.—In the use of the tables for propellers it must be borne in mind that they apply to carefully made propellers, with true smooth surfaces and sharp edges. If any of these features are lacking, allowance must be made, which can best be done by comparison of results from such propellers with the known properties of the experimental propellers.

Degree of Accuracy.—The degree of accuracy to be attributed to Taylor's experiments has already been stated to be somewhat better than one per cent in power; and as the power varies as the square of the diameter the diameter factors may conversely be given an accuracy of about one-half of one per cent for the model experiments. But attention has been called to the possible inaccuracy of the law of similitude as applying to propellers, which may amount to one or two per cent when the large propellers are made with the care and precision of the models. Rough, blunt-edged propellers may absorb somewhat more power than well-made propellers; they will show a marked loss of efficiency in some cases of three to five per cent or more.

The degree of precision of one per cent or better is to be attributed to those parts of the tables which are derived directly from Taylor's experiments; but certain parts of the tables have been extrapolated and are subject to more uncertainty, amounting perhaps, to two per cent. This reservation applies to the upper left-hand corner of the four-bladed table area-ratio 0.28, and the three-bladed table area-ratio 0.21, and to the greater part of four-bladed table area-ratio 0.72, and three-bladed table area-ratio 0.54.

Characteristics.—The general characteristics of propellers, as shown by the tables, should be clearly held in mind by the designer.

In a given table, as, for example, that for three blades, area-ratio 0.27, it will be seen that the diameter factor and consequently the diameter is nearly constant for a given pitch-ratio, whatever

the slip may be. There is some variation, usually a decrease as the slip increases, followed by an increase; thus at pitch-ratio 1.2 the values of D and the efficiency vary as follows:

Slip	0.06	0.12	0.18	0.24	0.30
D	56.2	55.6	55.4	55.2	55.3
e	0.693	0.710	0.709	0.693	0.667

For a considerable range of slip the efficiency changes but little, but there is an appreciable falling off for large slips. These conditions vary somewhat for the various pitch-ratios.

The best efficiency for a given value of R will be found near the full-faced figures; in some cases a higher efficiency may be had for some other slip and pitch-ratio but corresponding to a different value of R. In order to take advantage of the higher efficiency it would be necessary to change the revolutions. For example, at pitch-ratio 1.2 in the table referred to, the best efficiency is found at slips 0.14 to 0.16, which correspond to $R=2.33$ to $R=2.43$, but a better condition for that range in R can be secured at pitch-ratio 1.4, slip 0.20.

The effect of area-ratio, that is, of width of blade, can be brought out by assembling values for the properties corresponding to a certain value of the revolution factor R. Thus in the three-bladed table at pitch-ratio 1.2 and near the full-faced figures we may select the following values:

THREE-BLADED PROPELLERS, PITCH-RATIO 1.2.

Area-ratio	0.21	0.27	0.36	0.45	0.54
Slip	0.20	0.22	0.22	0.22	0.24
R	2.70	2.74	2.68	2.67	2.70
D	54.4	55.3	55.9	56.4	56.2
e	0.693	0.700	0.699	0.678	0.647

The values of R are the tabular values, but as the value of D changes slowly those here set down can all be taken as corresponding to the initial value $R=2.70$. It will be seen that there is a slight increase in diameter as the area-ratio increases, and an appreciable loss of efficiency for wide blades. The whole effect is, however,

of secondary importance and we may conclude that the diameter required is practically the same for all widths of blade.

The effect of using two, three, or four blades can be brought out by the following abstract from tables having the same projected area-ratio per blade, all at pitch-ratio 1.2:

No. of Blades.	Area-ratio.	Slip.	R	D	e
4	0.36	0.22	2.93	52.9	0.668
3	0.27	0.22	2.74	55.3	0.700
2	0.18	0.22	2.66	57.9	0.716

Effect of Blade-thickness.—The tables for designing propellers are arranged to vary the thickness inversely as the width of the blade, as should be the case for sake of strength. The assigned thicknesses are likely to be minima except for small propellers, and may be required to be increased for large propellers and for those that deliver a relatively large thrust. The thickness of propellers designed from the tables may be increased to half again as much as that given, without appreciable effect. On the other hand, there is an appreciable gain in efficiency from reducing the thickness when this is possible, amounting to five per cent when the thickness ratio can be reduced to 0.02. This gain in efficiency is accompanied by a reduction in the power absorbed, so that there will be little if any reduction in the diameter of the propeller to drive a boat at a given speed.

Propellers designed by the tables will be but little affected by changes of thickness that occur in practice.

Comparison with Tables.—If the conditions of service of a ship are such that the tables for propellers cannot be used directly, they may still be used as a means of basing a design on the known performance of a ship of the same type.

The essential feature in the use of the table which cannot be determined directly from the trial of a ship is the wake, which is used for calculating the speed of advance of the propeller, in the equation,

$$V_a = (1 - w)V.$$

We may assume a probable wake and solve for R and D by the equations,

$$R=\frac{r^{\frac{1}{2}}(\text{S.H.P.})^{\frac{1}{4}}}{V_a^{\frac{5}{4}}},$$

$$D=\frac{dr^{\frac{2}{3}}}{(\text{S.H.P.})^{\frac{1}{6}}V_a^{\frac{1}{6}}},$$

the latter being from equation (25), page 82. These values may be compared with the proper table and if our first assumption of wake appears unsatisfactory we can try again.

In some cases it may appear desirable to increase (or decrease) the diameter factor D by a percentage in addition to seeking for a probable wake factor.

For example, a trial of the police-boat *Guardian* showed that the engine developed 530 indicated horse-power when making 138.6 revolutions per minute, the speed being 12.33 knots per hour. The diameter of the four-bladed propeller was 7.33 feet, the pitch-ratio was 1.5, the projected area-ratio about 0.4, and the apparent slip 0.18. Assuming a wake of 0.10, and a mechanical efficiency of 0.9, the values of R and D are

$$R=\frac{(138.6)^{\frac{1}{2}}(477)^{\frac{1}{4}}}{(11.1)^{\frac{5}{4}}}=2.70,$$

$$D=\frac{7.33(138.6)^{\frac{2}{3}}}{(477\times 11.1)^{\frac{1}{6}}}=47.0.$$

The area-ratio comes between 0.36 and 0.48; the comparison can be made with either table. The latter gives $R=2.66$ at pitch-ratio 1.5 and slip 0.28, at which $D=48.2$.

Now since the apparent slip from the trial was 0.18, and the nearest tabular value is 0.28 the wake factor chosen appears to be too small. Solving equation (13), page 74, for wake,

$$1-w=(1-s)\div(1-s_1)=0.88, \qquad w=0.12.$$

A repetition of this work with 0.12 for the wake factor gives but a slight improvement. So we may conclude that the wake may be taken as 0.10 or 0.12.

Now the diameter is directly proportional to the diameter factor D, and as the computed result is about two per cent smaller than the tabular value, we may further allow for peculiarities of the propeller by subtracting two per cent from the value which we get from the use of the tables.

The propeller was of cast iron with rather blunt edges and an unfinished surface.

For example, the torpedo-boat *Biddle* on trial developed 4225 indicated horse-power on two screws, making 325.2 revolutions per minute and had a speed of 30 knots and an apparent slip of 0.142. The diameter of the propellers was 6.68 feet, the pitch-ratio 1.63, and the projected area-ratio about 0.59.

An assumption of zero wake gives $R=1.70$, and turning to the three-bladed table for area-ratio 0.54, this comes for a pitch-ratio of 1.63 at about 0.13 slip. The corresponding value of D is 51.0, which is in close concordance with the tabular value.

Tow-boat Propellers.—The conditions of service of a tow-boat are peculiar and incompatible; running free the speed is fairly high, 12 to 14 knots per hour; when towing the speed may be half or less as much as when running free. A part of the duty is to push large ships into position, the speed being then practically zero. Tow-boats are relatively short, and the water lines may be fairly full, but there is a good rise of floor, so that the block-coefficient is low. The pitch-ratio of the propeller is about 1.5, and the apparent slip running free may be about 0.20. The slip when towing is likely to be 0.50 or more; when pushing a ship into position the slip is nearly unity.

The propellers are four-bladed, made of cast iron, with straight edges, and wide tips; the projected area-ratio is large. From Froude's tests it appears that propellers with wide tips take about the same power as those with oval contours, but that the efficiency is two per cent less, or smaller. The propeller tables for four blades and large area-ratios may be used directly or may be made the basis of comparison with good practice by the method just given. Little is known about towing, consequently the design is made for running free.

Though deviating from common practice it is recommended

that the rounded form of the standard projected contour be used for tow-boats, and that the projected area be not made excessive; an area-ratio of 0.48 or 0.60 will be found sufficient. If a quicker running engine can be used better results will be obtained for towing from the use of a moderate pitch-ratio, not more than unity.

Steam-launches, especially for serving war-ships, have some of the characteristics of tow-boats and may be designed in the same way, except that the towing speed is relatively higher and the area-ratio need not be so high.

Small-boat Propellers.—The owner or prospective purchaser of a small boat often is confronted with the questions, what engine should be selected and what propeller should be chosen to go with the engine? Knowing the length and beam of his boat, the engine may be selected by aid of Keith's method on page 28, which allows the determination of the speed approximately.

Unless there is reason to the contrary the propeller may have three blades and a projected area-ratio of 0.27, that is, the table on page 117 may be used. The wake can be assumed to be $w=0.1$, except for racers which are likely to have zero wake.

Problem.—Required the propeller for a boat to make 7 knots per hour with a 10 horse-power engine which runs at 450 revolutions per minute. This corresponds to the problem on page 28, where it is computed that a 10 horse-power engine will give a speed of 7 knots to a boat that is 32 feet long and has a beam of $8\frac{1}{2}$ feet. This being a cruiser it may be assigned a wake of ten per cent, $w=0.10$; consequently the speed of advance will be

$$V_a=(1-w)V=(1-0.10)7=6.3 \text{ knots.}$$

Equation (23) on page 81 gives for the revolution factor,

$$R=\frac{r^{\frac{1}{2}}(\text{S.H.P.})^{\frac{1}{4}}}{V_a^{\frac{5}{4}}}=\frac{\overline{450}^{\frac{1}{2}}\times\overline{10}^{\frac{1}{4}}}{\overline{6.3}^{\frac{5}{4}}}=\frac{21.2\times1.78}{10}=3.77.$$

The various powers required are interpolated in the tables on pages 122 to 125. Having R we turn to page 117 for three-bladed

propellers and find at pitch-ratio unity and real slip 0.30 the value $D=59.2$ corresponding to $R=3.78$; the efficiency is 0.64. Equation (25), page 82, gives for the diameter,

$$d=\frac{D(\text{S.H.P.})^{\frac{1}{6}}(V_a)^{\frac{1}{6}}}{r^{\frac{2}{3}}}=\frac{59.2\times\overline{10}^{\frac{1}{6}}\times\overline{6.3}^{\frac{1}{6}}}{\overline{450}^{\frac{2}{3}}}=\frac{59.2\times1.47\times1.36}{58.7}=2\text{ ft.}$$

Bow-screws.—Screws are properly placed at the stern so that the wake gain may offset the thrust-deduction. A bow-screw throws a stream of water against the bow and produces an augmentation of resistance, and further it reduces the wake for the stern screw. The only ships with bow-screws are double-ended ferry-boats, and for them Col. E. A. Stevens advises that the propellers be so designed that the stern screw shall be as efficient as possible and that it be depended on for driving the boat. The forward screw should be inefficient, in fact, it should act as little as possible. His experience is that a blade with the thickness applied to the back and with blades raked away from the hull will conform to requirements both as the stern screw (driving) and the bow-screw (idle). It does not appear certain whether the rake of the blade is essential, though it is known that blades raked forward are inefficient. Perhaps the most effective way of making the bow-screw run idle would be to give it zero virtual slip in that position This could best be accomplished from model experiments, but a fair approximation can be had by dealing with the medial line (see Fig. 29, page 64) at about the mid-length of the blade and providing that the pitch of this line at the following edge shall give no slip when the screw acts as a bow-screw.

Number of Propellers.—Though there is little direct information on the subject it is probable that single screws are more efficient than twin screws, and that there is a progressive disadvantage in using triple and quadruple screws. The differences are not large and any type under favorable conditions may be more efficient than others for which favorable conditions cannot be secured. A single engine is, of course, simpler and cheaper than two engines with the same power, and in like manner two are cheaper than three or four. For moderate powers and speeds a

single screw will be chosen unless there are distinct advantages such as handiness or greater security from breakdown, which justify the greater expense. For example, all war-ships have two screws or more; and turbine steamers have two, three, or four screws for the better accommodation of the turbines. Large ships and high-speed ships most commonly have twin-screws to get favorable conditions for designing them.

Inclination of Shafts.—In a general way the flow of water at the stern of a ship is upward and inward, that is, toward the middle line. In order to get a flow parallel to the shaft of the propeller, the shaft should be inclined in the same way. In a few cases a shaft has been inclined upward in order to get the engine lower down. Cable-laying steamers have had their twin-shafts inclined in to get better maneuvering. But generally any inclination of the shaft has been in the wrong direction either to get better immersion for a single screw or to spread twin screws.

Now the effect of the flow of an inclined stream past the propeller is to vary the slip and consequently the thrust of a given blade. The angle which the blade makes with the stream flowing past it is always small, 5° being a fair estimate. It will therefore appear that inclinations of the shaft outward or downward are to be avoided, and that only small inclinations in such directions should ever be allowed. The effect of inclination of flow from the line of the shaft is to reduce the efficiency and to cause vibrations. The effect on efficiency is not known further than that propellers in towing-tanks show as good an efficiency behind a model as they do in open water, but this is not conclusive even for models. It is but too well known that propellers and especially turbine propellers cause unpleasant vibrations. As such propellers are set well clear of the hull it may be fair to charge the vibration in part to inclination of flow. Some large turbine steamers with four screws have had the out-board screws changed from three to four blades.

Cavitation.—When an attempt is made to apply an excessive power to a quick-running propeller, the stream of water acted on appears to break into eddies and the propeller cannot absorb the power or deliver the thrust expected. This phenomenon appears

to have been first identified by Mr. S. W. Barnaby on the torpedo-boat destroyer *Daring*, and was called cavitation by him. The propellers which showed this failure were of the Admiralty type with a width about 0.2 of the diameter. After the blades were made half again as wide and the pitch slightly increased the boat made 29 knots, then an unprecedented speed.

Mr. Barnaby concluded that the phenomenon was due to an attempt to produce too large a thrust for the area of the blades. Having computed the mean thrust per square inch of the projected blade area, he found that the stream broke when that pressure became $11\frac{1}{4}$ pounds, and that the difficulty was remedied by increasing the area so as to avoid so large a thrust. He further concluded that deeper immersion of the propeller would allow somewhat greater mean thrust. Since that time Mr. Barnaby has used his method with satisfaction for high-speed ships including turbine steamers.

In a paper on the application of steam turbines to ship propulsion Mr. E. M. Speakman quoted the performance of a number of steamers, giving among other things the thrust per square inch of projected area and the peripheral speed of the tips of the blades. He expressed the opinion that cavitation is liable to occur when the thrust exceeds 12 pounds per square inch or when the peripheral speed exceeds 12,000 feet per minute.

Unfortunately cavitation cannot be produced in the towing-tank for normal propellers, and those instances in which it has inadvertently occurred in practice have not been reported in such a way as to form a satisfactory basis for a theory.

Having made the blades as thin and sharp as possible it will be wise to restrict the peripheral speed to 12,000 feet per minute and to limit the thrust per square inch by Mr. Barnaby's method to 12 or 14 pounds per square inch.

To compute the thrust per square inch we may first find the effective horse-power by multiplying the indicated horse-power by the coefficient of propulsion—from 0.5 to 0.65. The effective horse-power may be multiplied by 33,000 to find the foot-pounds per minute, and this quantity divided by the speed of the ship in feet per minute ($101.3V$) will give the tow-rope resistance; this

last quantity must be divided by $1-t$ to find the thrust of the propeller; so that

$$\text{Thrust}=\frac{33000 \text{ E.H.P.}}{101.3V(1-t)},$$

in which V is the speed of the ship in knots and t is the thrust-deduction (about 0.1).

The total thrust is to be divided by the allowable thrust to find the projected area of all the blades; or conversely we may divide by the projected area to find the thrust per square inch. Precision is not important in this matter.

Example.—Let it be required to investigate the propellers for a turbine steamer that has a speed of 20 knots per hour, and a shaft horse-power of 10,500, applied to three screws. The propellers have a diameter of $6\frac{2}{3}$ feet and make 450 revolutions per minute.

Assuming a coefficient of propulsion from the shaft horse-power of 0.6, the effective power per screw will be

$$10500\times 0.6\div 3=2100.$$

If the thrust-deduction is assumed to be 0.1, the thrust will be

$$\frac{33000\times 2100}{101.3\times 20\times 0.9}=38000 \text{ pounds.}$$

A circle $6\frac{2}{3}$ feet or 80 inches in diameter has an area of 5026 square inches, and if the area-ratio per blade is 0.20, the area for three blades will be

$$5026\times 0.2\times 3=3015 \text{ square inches;}$$

and the thrust per square inch will be

$$3800\div 3015=12.6 \text{ pounds.}$$

A circle $6\frac{2}{3}$ feet in diameter has a perimeter of 20.9 feet, so that the peripheral speed of the tips of the blades will be

$$450\times 20.9=9400 \text{ feet per minute.}$$

Theory of Mechanical Similitude.—The conceptions of geometrical similitude and some of the simpler conclusions from the theory of mechanical similitude are so embedded in practical engineering that the extensions to the cases quoted in this book will probably be accepted by the casual reader without much hesitation. In the presentation of a method for practical use rather than for technical training, it was thought best to count on such an acceptance of the rules of similitude and to reserve a statement of the theory for those who have leisure and interest for it. More especially as the statement of the theory requires a careful definition of the fundamental conceptions of mechanics.

Velocity.—The rate of motion of a body is known as the velocity; if the body moves uniformly, the velocity can be found by dividing the space passed over by the time required to pass over it. If the velocity is not uniform, the velocity is found by taking a small distance along the path and dividing by the small time required.

Acceleration.—The rate of increase of velocity is known as acceleration. If the rate is uniform the acceleration can be found by dividing the increase in velocity by the time required. If the acceleration is not uniform it can be found by taking a small increase in velocity and dividing by the small increase in time.

Force.—The weight of a body is the force with which gravity attracts it toward the earth. Statical forces can be measured directly or indirectly by comparing with the weight of a standard piece of metal; moving forces cannot be so measured but are determined by comparison with the acceleration produced by gravity.

To be precise we first determine the mass of a body by measuring the acceleration produced by gravity on a piece of metal at a certain place; the actual experiments are not so simple, but that is a matter of detail. The mass of the body is now computed by the equation,

$$\text{Mass} = \frac{\text{weight}}{\text{acceleration}}, \quad \text{or} \quad m = \frac{w}{g} = \frac{w}{32.16},$$

where g is taken as the mean acceleration of gravity at the surface of the earth.

One of the fundamental conceptions of mass is that it is invariable, although weight and acceleration vary from place to place.

If some other force than gravity acts on a body to produce velocity it can be measured by the equation,

$$\text{Force} = \text{mass} \times \text{acceleration}, \quad \text{or} \quad f = ma.$$

Table for Mechanical Similitude.—There is given below a table for mechanical similitude giving the functions to which various properties are proportional.

In this table the fundamental units are those of length, time, and mass.

Geometrically the areas of similar figures are proportional to the square of a linear dimension and the volumes are proportional to the cube of a linear dimension.

TABLE FOR MECHANICAL SIMILITUDE.

Properties.	Symbols.	Functions.
Linear dimension.....	l	
Time................	t	
Mass................	m	
Surface.............	A	l^2
Volume..............	V	l^3
Velocity............	v	$\frac{l}{t}$
Acceleration..........	a	$\frac{v}{t} \propto \frac{l}{t^2}$
Force................	f	$ma \propto \frac{ml}{t^2}$
Work................	W	$fl \propto \frac{ml^2}{t^2}$
Power...............	P	$\frac{W}{t} \propto \frac{ml^2}{t^3}$
Density.............	d	$\frac{f}{V} \propto \frac{m}{l^2t^2}$

The definition of velocity gives at once the form of the function $\frac{l}{t}$, which may be read as the length or space passed over divided by the time required.

In like manner the first form of the function for the acceleration comes from the definition; the second form is obtained by substituting the function for the velocity. The second form is correctly written as proportional to the first; it is not equal for a numerical factor must be introduced which is $\frac{1}{2}$ for uniform acceleration.

The measurement of force is represented by the function *ma*; the second form introduces the quantity which is proportional to the acceleration.

Work is defined as the product of a force by the distance through which it acts. This gives the first form of the function, in the table, and the second, is obtained by introducing the proportional function for the force.

Power is the rate of doing work and is expressed by dividing the work by the time in which it is done. The second form of the function introduces the proportional function for force.

Density is the weight per unit of volume obtained by dividing the total weight (or force) by the volume; which latter is proportional to the cube of a linear dimension. The linear dimension in the proportional function for force reduces l in the denominator to the square.

In dealing with propulsion of ships the density of the water is constant which gives

$$\frac{f}{V}=d=\text{constant},$$

and the force (which is here weight or displacement in tons) is proportional to the volume, so that

$$D \propto V \propto l^3,$$

as has been assumed in the discussion of power.

Relative Speed.—The condition of relative speed comes from the assumption that the resistance shall be proportional to the displacement, that is,

$$R \propto D \propto l^3.$$

Remembering that resistance is a force and using the proportional function,

$$\frac{ml}{t^2} \propto l^3.$$

But at a given place the mass is proportional to the weight or displacement which has been shown to be proportional to l^3, so that the above proportion can be reduced to

$$\frac{l^2}{t^2} \propto l;$$

or remembering that the first member is the proportional function for velocity,

$$v^2 \propto l.$$

Writing this in the form of a proportion with V to the first power to represent the speed of the ship in knots per hour,

$$V_1 : V_2 :: \sqrt{L_1} : \sqrt{L_2};$$

where the linear dimension chosen is the length of the ship in feet.

This is the proportion of relative speeds; and these are the speeds at which the resistances are proportional to the displacements.

Extended Law of Comparison.—The proportional function for power gives

$$P \propto \frac{ml^2}{t^3} = l^2 \frac{l^3}{t^3}.$$

Replacing $\frac{l^3}{t^3}$ by v^3 from the proportional function for velocity, we have

$$P \propto l^2 v^3,$$

but the relative velocity is proportional to the square root of a linear dimension, so that

$$P \propto l^2 l^{\frac{3}{2}} = l^{\frac{7}{2}};$$

or, writing the above in the form of a proportion with indicated horse-power and the length of the ship,

$$(\text{I.H.P.})_1 : (\text{I.H.P.})_2 :: L_1^{\frac{7}{2}} : L_2^{\frac{7}{2}}.$$

Since the displacement is proportional to the cube of a linear dimension the proportion may be

$$(\text{I.H.P.})_1 : (\text{I.H.P.})_2 :: D_1^{\frac{7}{6}} : D_2^{\frac{7}{6}}.$$

Sometimes the shaft horse-power (S.H.P.) is used instead of the indicated horse-power.

Admiralty Coefficient.—To show that the method of the Admiralty coefficient is a variant of the extended law of comparison, the velocity is made proportional to the square root of a linear dimension for then

$$\text{I.H.P.} = \frac{D^{\frac{2}{3}} V^3}{K} \propto l^2 l^{\frac{3}{2}} = l^{\frac{7}{2}}.$$

Independent Estimate.—Of the two parts that enter into the independent estimate of power the second dependent on the wave-making resistance conforms to the laws of similitude, but the first, dependent on the surface friction, does not. The power to overcome wave-making resistance has the form

$$0.00307\, b \frac{D^{\frac{2}{3}}}{L} V^5 \propto \frac{l^2}{l} l^{\frac{5}{2}} = l^{\frac{7}{2}}.$$

The power to overcome frictional resistance has the form,

$$0.00307 f S V^{n+1},$$

where n is less than two. If n were two the form would conform to the law of similitude because then we would have

$$SV^3 \propto l^2 l^{\frac{3}{2}} = l^{\frac{7}{2}}.$$

Since the experiments of Froude show conclusively that the resistance of friction increases with a power of the speed less than two, it is clear that the theory of similitude tends to overestimate power for a larger vessel than the type, for speed-length-ratios less than unity.

If we consider the entire equation for the independent extimate

$$0.00307\left(fSV^{n+1} + b\frac{D^{\frac{2}{3}}}{L}V^5\right),$$

it appears that the first term increases as a power of V less than the cube, while the second term increases as the fifth power. So long as the first term is preponderent, the combined influence of both terms may make the power increase as the cube of the speed, as is assumed by the Admiralty coefficient.

For speed-length-ratios which approach unity, the second term has large influence and the power increases faster than would be indicated by the cube of the speed. If the speed-length-ratio is greater than unity the exponent of the speed may be four or even larger.

Keith's Method.—The equation for finding speed of small boats on page 28 may be reduced as follows:

$$V = C\frac{\sqrt[3]{LP}}{B} \propto \frac{l^{\frac{1}{3}}(l^{\frac{7}{2}})^{\frac{1}{3}}}{l} = l^{\frac{1}{2}},$$

which agrees with the condition for corresponding speed.

Revolutions of Propeller.—From equation (8) on page 64 we have

$$pr(1 - s) = 101.3V_a,$$

where p is the pitch of the propeller in feet, r represents the revolutions per minute and s is the real slip while V_a is the speed of

advance of the propeller. If the slip is assumed to be constant, then

$$r \propto \frac{V_a}{p},$$

and since the pitch is a linear dimension and the speed varies as the square root of a linear dimension, we have

$$r \propto \frac{1}{\sqrt{l}}.$$

Writing this as a proportion we have

$$r_s : r_m :: \frac{1}{\sqrt{l_s}} : \frac{1}{\sqrt{l_m}},$$

which is the proper proportion for the revolutions per minute of the propellers of a ship and its model. Then if the model is one-sixteenth as long as the ship its propeller should make four times as many revolutions per minute. This relation does not hold in passing from a type ship to a new design, for in that case the number of revolutions depends on other conditions; for reciprocating engines the piston speed is usually constant, which, for a larger ship requires fewer revolutions than the above proportion would indicate.

Propeller Equations.—For the propeller equation on pages 81 and 82, we may readily show conformity with the theory of similitude now that the revolutions are found to vary inversely as the square root of a linear dimension. As for equation (23), we have

$$R = \frac{r^{\frac{1}{2}}(\text{S.H.P.})^{\frac{1}{4}}}{V_a^{\frac{5}{4}}} \propto \frac{(l^{\frac{7}{2}})^{\frac{1}{4}}}{(l^{\frac{1}{2}})^{\frac{1}{2}}(l^{\frac{1}{2}})^{\frac{5}{4}}} \propto \frac{l^{\frac{7}{8}}}{l^{\frac{7}{8}}},$$

that is R is a numerical factor independent of the size of the propeller.

Again equation (25) is

$$d = \frac{D(\text{S.H.P.})^{\frac{1}{6}} V_a^{\frac{1}{6}}}{r^{\frac{2}{3}}} \propto (l^{\frac{7}{2}})^{\frac{1}{6}}(l^{\frac{1}{2}})^{\frac{1}{6}}(l^{\frac{1}{2}})^{\frac{2}{3}} = l,$$

for here D is a numerical factor; the diameter therefore is correctly proportional to a linear dimension.

Engine Power and Weight.—The power of a steam emgine is computed by the equation

$$\text{I.H.P.} = 2\frac{pasr}{33000},$$

in which p is the mean effective pressure as determined by the indicator, a is the area of the piston in square inches, and s is the stroke in feet, while r is the revolutions per minute.

For a given type of engine the steam pressure and the piston speed are likely to be the same, independent of the size; meaning by the piston speed the quantity,

$$2sr = \text{constant}.$$

This condition requires that the revolutions of an engine shall be inversely proportional to the stroke. The power of the engine, from the equations above, becomes proportional to

$$asr \propto d^2 s \frac{1}{s} \propto d^2,$$

where d is the diameter of the cylinder; that is, to the square of a linear dimension. We may therefore write the proportion,

$$(\text{I.H.P.})_1 : (\text{I.H.P.})_2 :: d_1^2 : d_2^2.$$

If the engines are of similar construction the weights will be proportional to the cube of a linear dimension, so that

$$W_1 : W_2 :: d_1^3 : d_2^3 :: (\text{I.H.P.})_1^{\frac{3}{2}} : (\text{I.H.P.})_2^{\frac{3}{2}}.$$

But the theory of mechanical similitude makes the indicated horse-power for a ship proportional to the seven-sixths power of the displacement, so that the ordinary convention that the piston speed shall be constant leads to the proportion,

$$W_1 : W_2 :: D_1^{\frac{7}{4}} : D_2^{\frac{7}{4}}.$$

This shows clearly the difficulty or impossibility of attaining relatively high speeds with large ships.

It is worthy of note that the weight of the engine increases faster than the power even when the revolutions are made inversely as the square root of the length, as required by the theory of similitude. In this case the equation for indicated horse-power gives the proportion,

$$(\text{I.H.P.})_1 : (\text{I.H.P.})_2 :: a_1 s_1 \frac{1}{\sqrt{L_1}} : a_2 s_2 \frac{1}{\sqrt{L_2}}.$$

Replacing the horse-power by the seven-halves power of a linear dimension and transferring the $\sqrt{L}$ from the second ratio to the first, we have

$$L_1^4 : L_2^4 :: d_1^3 : d_2^3 :: W_1 : W_2;$$

so that

$$W_1 : W_2 :: D_1^{\frac{4}{3}} : D_2^{\frac{4}{3}}.$$

For sake of comparison we may reduce to a common denominator and it appears that while the theory of similitude makes the power increase as the $\frac{14}{12}$ power of the displacement, it demands that the weight of the engine shall increase as the $\frac{16}{12}$ power; on the other hand, constant piston speed makes the engine weight increase as the $\frac{21}{12}$ power of the displacement.

This discussion applies only to the engine and not to the boilers, the weight of which for a given type may be expected to be proportional to the power.

In the development of marine engineering improvements in construction and design have much reduced weight of types of engines so that the conclusions just given will apply only to a given type and period.

Internal Propulsion.—A motor or an engine in a vessel can propel it only by acting on the water in which it floats, and must impress a sternward velocity on the water affected. Suppose that the propeller acts on W pounds and imparts to it an acceleration of

a feet per second. Then the force which is exerted on the water will be equal to the mass multiplied by the acceleration, or,

$$\frac{Wa}{g},$$

where g is the acceleration due to gravity.

If the ship has a speed of v feet per second, the work done per second will be

$$\frac{Wav}{g}.$$

The kinetic energy imparted to the stream of water set in motion will be

$$\frac{Wa^2}{2g}$$

per second. The total work is the sum of the two amounts, while the first part is the useful work. Consequently, the efficiency may be considered to be

$$e=\frac{\frac{1}{g}Wav}{\frac{1}{g}Wav+\frac{1}{2g}Wa^2}=\frac{v}{v+\frac{1}{2}a}.$$

If this is to be applied to the propulsion of a ship by a propeller we may transform the expression for efficiency as follows:

$$e=\frac{1}{1+\frac{a}{2v}}=\frac{1}{1+\frac{prs}{2pr(1-s_1)}}=\frac{1}{1+\frac{s}{2(1-s_1)}}.$$

This transformation is made with the assumption that the acceleration of the water in feet per second is

$$prs \div 60$$

while the speed of the ship is

$$pr(1-s_1)\div 60$$

in feet per second; p and r are the pitch and revolutions of the propeller while s and s_1 are the real and apparent slips.

Example.—If the real slip is 0.25 while the apparent slip is 0.10, the efficiency by the above equation is

$$e=\frac{1}{1+\dfrac{0.25}{2(1-0.10)}}=0.88.$$

This is to be considered as a limit neglecting friction and other resistances. If a factor 0.8 be allowed for resistances the efficiency would appear to be

$$0.88\times 0.8=0.7$$

which is not unusual for high grade propellers. This method may be applied where direct information is lacking, to ship propellers or aeroplane propellers. For the latter s and s_1 are probably equal.

The form of equation (22) on page 80 may be justified by a special application of this method. In the first place the acceleration imparted to the water, may be assumed to be proportional to

$$sV_a$$

where s is the real slip and V_a is the velocity of advance of the propeller, the latter being taken as proportional to the speed of the ship. The mass of the water acted on may be made proportional to the area of the disk swept by the propeller blades and to the speed of advance, that is, to

$$d^2V_a.$$

The thrust of the propeller may therefore be made proportional to the product of the preceding quantities, that is, to

$$sd^2V_a^2.$$

The power may therefore be made proportional to this product and to the speed of the ship, or for the latter we may again substitute the speed of advance. Then

$$\text{S.H.P.} = \frac{A}{1000} d^2 V_a^3,$$

where A is a factor depending on the pitch-ratio, the real slip, and the form of the propeller; the factor 1000 is introduced to control the decimal point of A.

To show that this last equation conforms to the theory of similitude we may make V_a proportional to the square root of the length, whence as d is itself a linear dimension,

$$\text{S.H.P.} \propto l^2 l^{\frac{3}{2}} = l^{\frac{7}{2}},$$

as shown on page 100.

TABLES

FOUR-BLADED PROPELLERS.

PROJECTED AREA RATIO=0.28. THICKNESS RATIO=0.07.

Point off one place for D, two for R, Three for e.

Pitch Ratio.		Real Slip.														
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34
0.60	*R*	467	478	491	504	518	534	550	568							
	D	639	642	645	649	652	655	660	666							
	e	514	517	519	521	522	523	524	524							
0.65	*R*	421	432	444	458	472	488	504	521	538	557	576				
	D	620	622	624	627	630	633	636	640	645	650	655				
	e	533	536	539	541	542	542	543	543	542	542	540				
0.70	*R*	391	403	415	428	441	455	469	483	498	515	533	551	570		
	D	608	609	611	613	615	617	620	623	626	629	632	636	641		
	e	554	558	560	561	562	562	562	561	560	558	556	553	550		
0.75	*R*	365	376	388	400	412	425	438	452	466	482	499	516	535	**556**	578
	D	596	597	598	600	602	604	606	608	610	612	615	619	623	627	631
	e	568	574	578	580	581	581	580	578	576	574	570	566	561	556	550
0.80	*R*	339	349	360	372	385	398	412	426	440	456	472	488	**506**	526	548
	D	584	585	586	587	588	590	592	594	596	599	602	605	608	611	614
	e	584	590	594	595	596	595	593	591	589	586	582	577	572	567	562
0.90	*R*	301	310	319	329	340	352	365	378	392	406	422	**438**	456	476	495
	D	566	566	566	566	567	568	569	570	572	574	576	578	581	584	588
	e	607	611	615	618	619	620	618	616	612	608	604	598	592	585	578
1.00	*R*	272	280	288	297	307	317	329	342	355	368	**381**	395	411	429	449
	D	550	550	550	550	550	550	550	551	552	553	555	557	559	561	563
	e	628	632	635	637	638	638	636	633	630	626	621	616	609	601	592
1.10	*R*	250	257	264	271	280	290	300	312	325	**339**	353	368	383	399	416
	D	536	536	535	535	535	535	536	536	537	538	539	540	541	543	545
	e	649	652	653	654	655	654	652	649	645	639	633	626	618	610	600
1.20	*R*	227	234	241	248	256	266	277	289	**301**	313	326	340	354	368	384
	D	521	521	521	520	520	520	520	521	521	522	522	523	525	527	529
	e	654	669	673	676	678	677	674	670	663	656	647	638	629	620	611
1.30	*R*	209	215	222	229	236	245	254	**265**	276	288	300	312	325	339	354
	D	505	504	504	504	504	504	504	505	505	506	507	508	509	510	512
	e	672	679	683	685	686	687	685	682	676	669	660	650	640	630	621
1.40	*R*	195	200	206	212	219	226	**235**	245	255	266	278	291	304	317	332
	D	494	493	492	491	490	490	490	490	491	492	493	494	496	498	500
	e	680	685	689	692	694	695	694	691	686	678	669	659	649	639	628
1.50	*R*	182	187	193	199	206	**213**	221	230	239	250	261	272	284	296	309
	D	481	480	479	478	477	477	477	477	478	478	479	480	482	484	486
	e	683	690	694	697	699	700	698	695	691	684	676	668	658	648	638
1.60	*R*	170	175	181	187	**194**	201	208	217	226	235	245	255	266	278	290
	D	471	470	469	468	467	467	466	466	466	466	467	468	469	470	471
	e	678	685	691	696	699	700	699	697	694	689	682	673	663	653	643
1.80	*R*	153	158	164	**170**	176	183	189	197	205	214	223	232	242	252	263
	D	453	452	451	450	449	448	447	446	445	445	445	446	447	448	449
	e	671	681	688	692	696	698	697	695	692	688	681	673	664	654	644
2.00	*R*	140	145	**151**	156	162	169	176	183	190	198	206	215	224	234	245
	D	439	437	436	435	434	433	432	431	430	430	430	429	429	429	430
	e	668	676	683	688	691	692	692	691	688	683	676	669	660	650	640

FOUR-BLADED PROPELLERS.

PROJECTED AREA RATIO=0.36. THICKNESS RATIO=0.06.

Point off one place for *D*, two for *R*, three for *e*

Pitch Ratio.		Real Slip.														
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34
0.60	*R*	439	455	471	487	503	519	535	551	568	586					
	D	668	669	670	671	673	675	677	679	681	685					
	e	522	523	522	520	516	511	506	501	494	487					
0.65	*R*	406	420	434	449	464	480	496	512	529	547	565				
	D	653	653	654	654	655	656	657	659	661	664	667				
	e	543	546	548	548	547	545	541	537	532	527	522				
0.70	*R*	376	389	402	416	430	445	461	477	494	512	532	553	574		
	D	637	637	638	638	638	639	640	642	644	646	649	652	655		
	e	560	564	567	569	568	566	563	560	556	552	548	543	538		
0.75	*R*	350	362	374	387	400	414	429	445	462	480	498	517	537	559	
	D	624	624	623	623	623	624	625	626	627	629	631	633	636	639	
	e	575	581	584	586	586	584	581	578	574	570	565	560	555	548	
0.80	*R*	328	340	352	364	376	389	403	417	432	447	463	481	501	523	547
	D	610	610	610	609	609	609	610	610	611	612	614	616	618	621	624
	e	589	595	599	601	601	600	598	595	592	588	584	578	571	564	555
0.90	*R*	290	300	311	322	334	346	358	371	385	399	414	430	447	465	486
	D	598	598	597	597	596	596	596	596	596	596	597	598	600	602	604
	e	611	617	621	622	623	622	621	619	616	613	608	602	595	588	580
1.00	*R*	260	269	279	290	301	312	324	336	349	362	375	389	404	420	437
	D	570	569	567	566	566	567	565	565	565	565	566	566	567	568	568
	e	639	643	646	648	649	648	646	643	639	634	629	622	614	605	597
1.10	*R*	237	245	253	263	274	285	296	307	319	331	344	358	472	387	403
	D	553	552	551	550	549	548	547	546	546	546	546	546	547	547	548
	e	652	658	662	666	666	665	662	659	654	648	641	634	627	619	609
1.20	*R*	216	223	231	240	250	260	271	282	293	305	317	330	344	358	373
	D	538	536	534	532	531	530	530	529	529	528	528	528	529	530	531
	e	661	668	674	677	679	679	676	672	668	661	653	645	636	627	617
1.30	*R*	200	206	213	221	230	239	249	260	271	282	294	307	320	334	348
	D	523	521	519	518	517	516	515	514	513	512	512	512	512	513	514
	e	670	677	681	684	685	685	684	682	677	670	662	653	644	633	622
1.40	*R*	182	198	195	207	215	224	233	243	253	264	276	288	300	313	326
	D	511	509	507	505	503	501	499	498	497	496	496	496	496	497	499
	e	679	684	688	691	692	692	691	688	683	676	668	658	648	639	629
1.50	*R*	175	181	188	195	203	211	219	228	237	247	258	269	281	294	308
	D	500	497	495	493	491	489	487	485	484	483	483	482	482	483	483
	e	682	690	695	698	699	698	696	692	686	680	672	662	652	642	632
1.60	*R*	166	172	178	185	192	199	206	215	224	234	244	255	267	280	293
	D	490	486	483	482	479	477	475	474	472	471	470	469	469	469	470
	e	683	691	696	699	701	701	699	695	690	682	674	664	654	642	631
1.80	*R*	148	154	160	167	174	181	188	196	205	215	225	235	246	257	268
	D	471	467	464	461	458	455	453	451	450	418	447	446	445	445	446
	e	671	680	688	695	698	697	696	693	687	678	669	658	648	638	627
2.00	*R*	131	137	143	150	157	164	172	180	188	197	206	215	225	235	246
	D	452	448	444	441	438	435	433	431	429	427	426	425	424	423	423
	e	644	658	669	676	681	684	683	680	677	671	663	655	646	635	624

FOUR-BLADED PROPELLERS.

PROJECTED AREA RATIO=0.48. THICKNESS RATIO=0.05.

Point off one Place for *D*, two for *R*, three for *e*.

Pitch Ratio.		Real Slip.														
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34
0.60	*R*	397	410	424	440	456	473	492	511	530	550	570				
	D	716	714	712	711	709	707	705	704	703	701	700				
	e	513	519	524	528	530	531	531	530	529	528	524				
0.65	*R*	370	383	396	411	427	444	463	482	501	521	541	563			
	D	696	693	691	689	687	686	685	684	683	682	681	680			
	e	529	537	542	546	548	549	548	546	544	541	538	533			
0.70	*R*	345	358	372	487	402	418	435	452	470	490	510	531	552	575	
	D	676	673	671	670	669	668	666	665	664	663	662	662	661	660	
	e	541	550	557	561	564	566	564	562	560	556	551	545	539	531	
0.75	*R*	323	335	348	362	376	392	408	425	443	462	481	501	522	543	565
	D	659	656	654	652	650	649	648	647	646	645	645	645	644	644	644
	e	559	566	572	576	578	580	580	578	574	569	563	556	549	540	531
0.80	*R*	300	312	324	337	351	466	482	400	418	437	456	475	495	516	538
	D	643	640	637	634	632	631	630	630	630	630	629	629	629	630	630
	e	572	580	586	590	593	594	594	591	588	583	577	570	561	552	541
0.90	*R*	269	280	291	302	313	327	343	359	375	391	407	424	443	464	485
	D	618	614	610	607	605	603	601	600	599	598	598	598	599	600	600
	e	600	609	616	620	622	622	620	616	611	606	599	592	583	573	561
1.00	*R*	243	253	264	275	286	297	310	324	339	354	369	386	403	422	441
	D	598	593	589	586	583	580	577	575	573	572	572	571	571	572	573
	e	613	623	631	636	638	639	639	637	633	628	619	610	600	590	580
1.10	*R*	220	230	240	251	262	273	284	296	308	321	336	352	369	386	403
	D	580	574	569	566	563	560	557	554	552	550	549	548	548	548	548
	e	625	637	645	651	654	656	654	651	648	642	634	626	616	606	695
1.20	*R*	200	209	220	231	242	253	264	275	286	298	312	327	343	359	375
	D	561	557	552	548	544	541	539	537	535	534	532	531	531	530	530
	e	629	643	653	659	663	665	663	659	654	649	641	633	624	613	602
1.30	*R*	184	193	202	212	223	234	245	256	267	279	291	304	320	336	352
	D	548	542	538	533	528	524	522	520	518	517	516	515	514	513	512
	e	636	648	658	664	668	668	666	663	658	652	646	637	627	616	605
1.40	*R*	171	179	188	198	208	219	229	240	250	261	273	285	298	313	329
	D	535	529	523	517	514	510	507	504	502	500	499	498	497	497	496
	e	632	646	658	666	670	670	669	666	661	655	648	640	630	618	606
1.50	*R*	159	167	176	186	196	205	214	224	234	244	254	266	280	295	310
	D	522	515	510	505	500	496	492	489	488	486	484	482	481	480	479
	e	623	640	653	664	669	671	670	667	663	656	648	639	628	617	606
1.60	*R*	150	158	167	176	186	195	204	213	222	231	241	251	265	279	293
	D	512	505	498	492	488	484	480	477	474	472	470	468	467	466	466
	e	613	631	644	655	662	665	664	662	658	652	645	637	626	616	603
1.80	*R*	133	142	151	159	168	176	185	194	202	211	219	230	242	254	267
	D	493	485	478	472	466	462	459	456	453	451	449	447	445	444	443
	e	586	606	621	632	641	646	648	647	644	640	634	625	616	606	595
2.00	*R*	121	128	136	145	153	161	169	177	186	195	204	214	224	235	246
	D	476	467	459	453	448	444	440	436	433	431	429	428	427	426	426
	e	561	583	601	614	622	627	630	630	628	624	618	611	602	592	582

FOUR-BLADED PROPELLERS.

PROJECTED AREA RATIO=0.60. THICKNESS RATIO=0.04.

Point off one place for D, two for R, three for e.

Pitch Ratio.		Real Slip.														
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34
0.60	*R*	377	392	408	424	441	458	475	492	511	530	551	575			
	D	750	746	742	738	735	732	729	727	725	723	722	722			
	e	490	496	499	501	501	499	497	493	488	481	473	463			
0.65	*R*	347	362	377	392	408	424	441	458	477	497	517	539	563		
	D	724	719	715	712	709	706	703	701	700	699	699	698	698		
	e	502	508	513	516	518	519	517	513	508	502	493	483	471		
0.70	*R*	326	339	353	367	382	397	413	430	448	467	487	508	531	555	
	D	704	699	694	689	686	683	681	679	677	676	675	675	675	675	
	e	516	522	528	532	534	535	534	532	527	522	513	504	492	478	
0.75	*R*	305	318	331	445	459	374	390	406	423	441	460	480	500	522	547
	D	686	680	674	670	666	663	660	658	656	655	654	654	654	654	654
	e	525	535	541	545	548	550	550	548	544	539	531	523	512	500	487
0.80	*R*	286	299	312	325	339	354	370	386	403	420	437	456	476	497	520
	D	669	662	657	653	649	646	644	642	640	639	638	637	636	636	636
	e	535	545	553	558	562	564	565	564	561	556	548	540	531	520	508
0.90	*R*	255	268	281	294	307	321	335	350	365	380	395	412	431	450	470
	D	639	631	628	625	621	617	613	610	608	606	605	604	604	603	603
	e	544	562	573	681	587	590	590	588	585	580	574	566	557	547	536
1.00	*R*	231	243	255	268	281	294	307	320	334	349	364	379	396	413	431
	D	614	609	604	599	595	591	588	586	584	582	580	578	577	576	575
	e	553	572	586	597	603	606	608	607	605	600	594	585	576	566	555
1.10	*R*	211	222	234	246	258	270	282	294	308	322	336	351	366	382	399
	D	592	587	572	578	574	570	566	563	560	558	556	554	553	552	551
	e	556	577	594	606	614	618	621	621	618	613	606	597	588	578	566
1.20	*R*	192	203	214	225	236	248	260	272	285	298	312	326	341	357	373
	D	574	567	562	557	552	548	545	541	539	537	535	534	533	532	531
	e	556	578	596	607	618	621	624	624	621	616	611	605	595	585	574
1.30	*R*	177	188	199	210	220	231	242	253	264	277	290	303	316	331	347
	D	557	550	544	539	535	530	526	523	520	518	516	514	513	512	511
	e	554	571	597	608	615	620	623	624	622	617	611	605	596	586	577
1.40	*R*	166	176	186	196	206	216	226	236	247	259	271	283	295	310	326
	D	543	536	529	524	519	515	511	507	504	501	499	497	495	494	493
	e	549	579	599	611	617	623	622	621	618	614	610	604	597	589	579
1.50	*R*	155	165	174	184	194	203	213	223	234	245	256	267	280	294	309
	D	530	523	517	510	505	500	496	493	490	487	484	482	480	479	478
	e	539	571	594	608	616	619	619	618	615	610	605	599	592	583	574
1.60	*R*	145	154	163	173	183	192	202	212	221	231	241	253	266	279	293
	D	518	511	504	498	492	488	484	480	477	474	471	469	467	466	465
	e	524	558	582	596	605	611	612	611	608	605	600	593	586	578	570
1.80	*R*	130	139	148	156	165	173	182	191	201	211	221	233	245	257	269
	D	501	493	485	478	472	466	462	459	456	453	450	448	446	445	444
	e	502	540	561	576	584	589	591	591	589	586	582	576	568	561	553
2.00	*R*	118	126	135	143	152	160	168	176	185	195	204	215	227	238	250
	D	485	476	468	460	453	447	443	439	436	433	431	429	428	427	426
	e	486	523	545	559	566	570	571	570	568	565	560	555	548	541	533

FOUR-BLADED PROPELLERS.

PROJECTED AREA RATIO = 0.72. THICKNESS RATIO = 0.03.

Point off one place for D, two for R, three for e.

Pitch Ratio.		Real Slip.														
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34
0.60	*R*	366	378	393	408	424	441	459	477	497	517	538	561			
	D	762	756	750	746	741	738	736	733	731	730	729	729			
	e	441	446	449	453	456	459	461	463	464	464	461	455			
0.65	*R*	340	352	366	380	394	411	429	448	468	488	509	531	553	576	
	D	740	733	726	721	716	713	710	708	707	706	705	704	703	703	
	e	449	454	459	462	465	467	469	471	472	472	470	468	463	455	
0.70	*R*	319	330	343	356	370	385	401	418	437	457	478	500	522	545	571
	D	721	714	707	701	695	692	688	685	683	681	680	680	680	679	679
	e	461	466	471	475	477	479	481	482	482	481	479	476	472	465	456
0.75	*R*	298	310	323	336	349	362	377	393	411	430	451	473	495	518	542
	D	703	696	689	682	677	673	669	666	663	661	660	659	658	658	659
	e	469	476	481	486	490	492	493	494	493	491	489	485	480	473	464
0.80	*R*	280	291	303	316	329	342	357	373	388	405	425	446	468	491	515
	D	686	678	672	666	660	655	651	647	645	642	640	638	637	637	638
	e	473	485	494	501	504	506	507	508	507	505	502	498	491	484	477
0.90	*R*	250	261	273	285	298	310	322	336	351	367	385	404	425	448	471
	D	655	648	642	636	631	626	621	617	613	610	607	605	604	604	603
	e	476	499	515	525	531	534	535	535	534	532	528	524	517	509	501
1.00	*R*	224	234	246	258	270	281	293	306	319	334	351	369	388	400	413
	D	625	618	612	606	602	596	592	589	586	584	581	578	576	574	574
	e	465	495	522	540	551	556	559	560	560	558	553	547	539	529	518
1.10	*R*	205	215	225	236	248	260	272	284	296	308	323	340	358	377	397
	D	601	594	588	582	576	571	567	564	561	559	556	554	552	550	548
	e	461	496	523	542	556	564	571	574	575	574	571	566	556	546	536
1.20	*R*	189	199	209	220	231	242	253	265	277	289	302	316	333	351	369
	D	580	572	565	559	553	549	546	543	540	538	536	534	532	530	528
	e	460	497	525	546	560	570	576	580	581	580	578	572	564	554	542
1.30	*R*	180	189	198	208	218	229	240	250	261	272	284	297	312	328	345
	D	562	554	546	539	534	529	525	522	520	518	516	514	512	510	509
	e	463	500	528	549	564	574	579	581	582	580	577	572	565	556	546
1.40	*R*	168	177	187	197	207	217	227	238	249	260	271	282	295	310	326
	D	545	537	529	522	516	512	508	504	501	499	497	496	495	494	493
	e	456	496	524	546	561	571	576	579	578	576	572	568	562	554	544
1.50	*R*	160	169	178	187	197	207	217	227	238	249	260	271	282	295	310
	D	531	523	515	508	502	497	493	489	486	484	482	481	480	479	478
	e	446	486	514	536	551	562	568	570	570	568	564	560	555	548	539
1.60	*R*	153	161	170	179	189	199	209	219	229	239	250	261	272	283	296
	D	519	510	502	495	489	484	479	475	472	469	468	467	466	465	464
	e	430	469	496	522	539	550	556	560	561	560	556	552	548	540	531
1.80	*R*	142	150	159	168	177	186	196	205	215	225	235	246	256	267	278
	D	496	488	480	473	467	461	456	452	449	448	446	444	443	442	442
	e	395	435	463	488	504	517	525	529	531	532	530	526	521	514	507
2.00	*R*	132	140	148	157	166	175	185	195	204	214	225	235	246	256	267
	D	473	464	456	449	444	439	435	432	429	427	426	425	425	425	425
	e	347	389	419	439	456	470	481	487	492	494	495	493	490	485	478

THREE-BLADED PROPELLERS.

PROJECTED AREA RATIO=0.21. THICKNESS RATIO=0.07.

Point off one place for *D*, two for *R*, three for *e*.

Pitch Ratio.		Real Slip.														
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34
0.60	*R*	437	448	460	472	485	499	515	532							
	D	668	671	674	677	681	685	690	696							
	e	529	532	534	536	537	538	539	539							
0.65	*R*	394	404	416	428	442	457	472	488	504	521	539				
	D	648	650	652	655	658	661	665	669	674	679	685				
	e	548	551	554	556	557	558	559	559	558	557	555				
0.70	*R*	366	376	388	400	413	426	439	452	466	482	499	516	534		
	D	635	637	639	641	643	645	648	651	654	657	661	665	670		
	e	571	574	577	578	579	579	579	578	576	574	572	569	566		
0.75	*R*	340	350	362	374	386	398	410	423	436	451	467	483	501	520	540
	D	623	624	625	627	629	631	633	635	637	640	643	647	651	655	660
	e	585	591	595	597	598	598	597	595	593	591	587	583	578	573	566
0.80	*R*	317	326	337	349	361	373	386	399	412	427	442	457	474	493	512
	D	611	612	613	614	615	617	619	621	623	626	629	632	635	638	642
	e	602	608	612	613	614	613	611	609	607	604	600	595	590	585	579
0.90	*R*	282	290	299	308	318	330	342	354	367	380	395	410	427	445	463
	D	592	592	592	592	593	594	595	596	598	600	602	604	607	610	614
	e	626	630	634	637	638	639	637	635	631	627	623	617	610	603	596
1.00	*R*	255	262	270	278	287	297	308	320	332	344	357	370	385	402	420
	D	575	575	575	575	575	575	575	576	577	578	580	582	584	586	588
	e	648	652	655	657	658	658	656	653	650	646	641	635	628	620	611
1.10	*R*	234	240	247	254	262	271	281	292	304	317	330	344	358	373	389
	D	560	560	559	559	559	559	560	560	561	562	563	564	566	568	570
	e	671	673	675	676	677	676	674	671	666	660	654	647	639	630	620
1.20	*R*	213	219	225	232	240	249	259	270	281	293	305	318	331	344	358
	D	544	544	544	543	543	543	543	544	544	545	546	547	549	551	553
	e	687	692	696	699	701	700	697	693	686	678	669	660	651	642	633
1.30	*R*	195	201	207	214	221	229	238	248	258	269	280	292	304	317	331
	D	528	527	527	527	527	527	527	528	528	529	530	531	532	533	535
	e	696	703	707	709	710	711	709	706	700	692	683	673	663	653	643
1.40	*R*	182	187	192	198	205	212	220	229	239	249	260	272	284	297	311
	D	516	515	514	513	512	512	512	512	513	514	515	516	518	520	522
	e	705	710	714	717	719	720	719	716	711	703	693	683	673	662	651
1.50	*R*	169	174	180	186	192	199	207	215	224	234	244	254	265	277	289
	D	503	502	501	500	499	499	499	499	500	500	501	502	504	506	508
	e	709	716	721	724	726	727	725	722	717	710	702	693	683	673	662
1.60	*R*	159	164	170	176	182	188	195	203	211	220	229	239	249	260	271
	D	492	491	490	489	488	488	487	487	487	487	488	489	490	491	492
	e	706	713	719	724	727	728	727	725	722	717	709	700	690	680	669
1.80	*R*	142	147	153	159	165	171	177	184	192	200	208	217	226	236	246
	D	473	472	471	470	469	468	467	466	465	465	465	466	467	468	469
	e	700	710	717	722	726	728	727	725	722	717	710	702	693	683	673
2.00	*R*	131	136	141	146	152	158	164	171	178	185	193	201	210	219	229
	D	459	457	456	455	454	453	452	451	450	449	449	448	448	448	449
	e	700	709	716	721	724	725	725	724	721	716	709	701	692	682	671

THREE-BLADED PROPELLERS.

PROJECTED AREA RATIO=0.27. THICKNESS RATIO=0.06.

Point off one place for D, two for R, three for e.

Pitch Ratio.		Real Slip.														
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34
0.60	*R*	411	426	441	456	471	486	501	516	532	548					
	D	698	699	700	702	704	706	708	710	713	716					
	e	542	543	542	540	536	531	525	519	513	506					
0.65	*R*	380	393	406	420	434	449	464	479	495	511	527				
	D	682	682	683	683	684	685	687	689	691	694	697				
	e	565	568	570	570	569	567	563	559	554	549	544				
0.70	*R*	352	364	376	389	402	416	431	447	463	480	498	517	537		
	D	666	666	666	667	667	668	669	671	673	675	678	681	685		
	e	583	587	590	592	591	589	586	583	579	575	570	565	560		
0.75	*R*	328	339	350	362	374	387	401	416	432	449	466	484	503	523	
	D	652	652	651	651	651	652	653	654	655	657	659	662	665	668	
	e	599	605	608	610	610	608	605	602	598	594	589	584	578	571	
0.80	*R*	307	318	329	340	352	364	377	390	404	418	433	450	469	490	512
	D	638	638	638	637	637	637	638	638	639	640	642	644	646	649	652
	e	614	620	624	626	626	625	623	620	617	613	608	602	595	587	578
0.90	*R*	271	281	291	301	312	323	335	347	360	373	387	402	418	435	453
	D	625	625	624	624	623	623	623	623	623	623	624	625	627	629	631
	e	637	643	647	649	650	649	648	646	643	639	634	628	621	613	605
1.00	*R*	243	252	261	271	281	292	303	314	326	338	351	364	378	393	409
	D	596	594	593	592	591	591	590	590	590	590	590	591	592	593	594
	e	667	671	674	676	677	676	674	671	667	662	656	649	641	632	623
1.10	*R*	222	229	237	246	256	266	276	287	298	310	322	335	348	362	376
	D	578	577	576	575	574	573	572	571	571	571	571	571	572	572	572
	e	682	688	693	696	697	696	693	689	684	678	671	664	656	647	637
1.20	*R*	202	209	216	224	233	243	253	263	274	285	297	309	322	335	349
	D	562	560	558	556	555	554	554	553	553	552	552	552	553	554	555
	e	693	700	706	710	712	712	709	705	700	693	685	676	667	657	647
1.30	*R*	187	193	200	207	215	224	233	243	253	264	275	287	299	312	326
	D	547	545	543	541	540	539	538	537	536	535	535	535	535	536	537
	e	704	711	715	718	720	720	719	716	711	704	696	686	676	665	654
1.40	*R*	175	180	186	193	201	209	218	227	237	247	258	269	281	293	305
	D	534	532	530	528	526	524	522	521	520	519	518	518	519	520	521
	e	715	720	724	727	729	729	727	724	719	712	703	693	683	673	662
1.50	*R*	164	170	176	183	190	197	205	213	222	231	241	252	263	275	288
	D	523	520	517	515	513	511	509	507	506	505	505	504	504	505	505
	e	721	728	733	736	737	736	734	730	724	717	708	698	688	677	666
1.60	*R*	155	161	167	173	179	186	193	201	209	218	228	239	250	262	274
	D	512	508	505	503	501	499	497	495	493	492	491	490	490	490	491
	e	722	730	736	739	741	741	739	735	729	721	712	702	691	679	667
1.80	*R*	138	144	150	156	162	169	176	184	192	201	210	220	230	240	251
	D	492	488	485	482	479	476	474	472	470	468	467	466	465	465	466
	e	712	722	730	736	739	740	739	736	729	720	710	699	688	677	666
2.00	*R*	122	128	134	140	147	154	161	168	176	184	192	201	210	220	230
	D	472	468	464	461	458	455	453	451	449	447	445	444	443	442	442
	e	687	702	713	721	726	729	728	726	722	715	707	698	688	677	665

THREE-BLADED PROPELLERS.

PROJECTED AREA RATIO=0.36. THICKNESS RATIO=0.05.

Point off one place for *D*, two for *R*, three for *e*.

Pitch Ratio.		Real Slip.														
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34
0.60	*R*	371	383	397	412	427	443	460	478	496	515	534				
	D	749	747	745	743	741	739	737	736	735	733	731				
	e	542	548	554	558	560	561	561	560	559	557	554				
0.65	*R*	346	358	371	385	400	416	433	451	469	488	507	527			
	D	727	724	722	720	718	717	716	715	714	713	712	711			
	e	559	567	573	577	579	580	579	577	575	572	568	563			
0.70	*R*	323	335	348	362	376	391	407	423	440	458	477	497	517	538	
	D	707	704	702	700	699	698	696	695	694	693	692	692	691	690	
	e	573	582	589	594	597	599	597	595	592	588	583	577	570	562	
0.75	*R*	302	313	325	338	352	367	382	398	415	432	450	469	488	508	529
	D	689	686	684	682	680	678	677	676	675	674	674	674	673	673	673
	e	592	600	606	610	612	614	614	612	608	603	597	589	581	572	562
0.80	*R*	281	292	303	315	328	342	358	375	392	409	426	444	463	483	504
	D	672	669	666	663	661	660	659	658	658	658	657	657	657	658	658
	e	606	615	621	626	629	630	630	627	623	618	612	604	595	585	574
0.90	*R*	252	262	272	282	293	306	321	336	351	366	381	397	415	434	454
	D	646	642	638	635	632	630	628	627	626	625	625	625	626	627	627
	e	637	647	654	658	660	660	658	654	649	643	636	628	619	608	595
1.00	*R*	227	237	247	257	267	278	290	303	317	331	345	361	377	395	413
	D	625	620	616	612	609	606	603	601	599	598	598	597	597	598	599
	e	652	663	671	676	679	680	680	678	674	668	659	649	639	628	615
1.10	*R*	206	215	225	235	245	255	266	277	288	300	314	329	345	361	377
	D	606	600	595	591	588	585	582	579	577	575	574	573	573	573	573
	e	666	679	688	694	697	699	697	694	690	684	676	667	657	646	634
1.20	*R*	187	196	206	216	226	236	247	258	269	280	292	306	321	336	351
	D	587	582	577	573	569	566	563	561	559	588	556	555	555	554	554
	e	672	687	697	704	708	710	708	704	699	693	685	676	666	655	643
1.30	*R*	172	180	189	199	209	219	229	239	250	261	272	284	299	314	329
	D	573	567	562	557	552	548	546	544	542	540	539	538	537	536	535
	e	682	695	705	712	716	716	714	710	705	699	692	683	672	660	648
1.40	*R*	160	167	176	186	195	205	214	224	234	244	255	266	279	293	308
	D	559	553	547	541	537	533	530	527	525	523	522	521	520	519	518
	e	680	695	707	716	721	721	719	716	711	704	697	688	677	664	651
1.50	*R*	149	157	165	174	183	192	201	210	219	228	238	249	262	276	290
	D	546	539	533	528	523	519	515	512	510	508	506	504	503	502	501
	e	672	690	704	716	722	724	723	720	715	708	700	689	676	666	654
1.60	*R*	140	148	156	165	174	182	191	199	207	216	225	235	248	261	274
	D	535	528	521	515	510	506	502	499	496	494	492	490	488	487	487
	e	663	683	697	709	716	719	718	716	712	706	698	689	678	666	653
1.80	*R*	125	133	141	149	157	165	173	181	189	197	205	215	226	238	250
	D	515	507	500	493	487	483	480	476	473	471	469	467	465	464	463
	e	642	660	676	689	698	704	706	705	702	697	690	681	671	660	648
2.00	*R*	113	120	128	135	143	150	158	166	174	182	191	200	210	220	230
	D	497	488	480	474	468	464	460	456	453	451	449	447	446	445	445
	e	615	639	659	673	682	687	690	690	688	684	678	670	660	649	638

THREE-BLADED PROPELLERS.

PROJECTED AREA RATIO = 0.45. THICKNESS RATIO = 0.04.

Point off one place for D, two for R, three for e.

Pitch Ratio.		Real Slip.														
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34
0.60	*R*	353	367	382	397	413	429	445	461	478	496	516	538			
	D	784	780	776	772	768	765	762	760	758	756	755	754			
	e	527	533	537	539	539	537	534	530	524	517	508	498			
0.65	*R*	325	339	353	367	382	397	413	429	447	465	484	505	527		
	D	757	752	748	744	741	738	735	733	731	730	730	729	729		
	e	540	547	552	555	557	558	556	552	547	540	531	520	507		
0.70	*R*	305	317	330	343	357	371	386	402	419	437	456	476	497	519	
	D	736	730	725	721	717	714	712	710	708	707	706	705	705	705	
	e	556	563	569	573	575	576	575	573	568	562	553	543	530	515	
0.75	*R*	285	297	310	323	336	350	365	380	396	413	430	449	468	489	512
	D	717	711	705	700	696	693	690	688	686	685	684	684	684	684	684
	e	567	577	584	588	591	593	593	591	587	581	573	564	553	540	525
0.80	*R*	268	280	292	304	317	331	346	361	377	393	409	427	445	465	486
	D	699	692	687	682	678	675	673	671	669	668	667	666	665	665	665
	e	578	589	597	603	607	609	610	609	606	600	593	584	573	561	548
0.90	*R*	239	251	263	275	287	300	313	327	341	355	370	386	403	421	440
	D	668	660	657	653	649	645	641	638	636	634	632	631	631	630	630
	e	589	608	620	629	635	638	639	637	633	628	621	613	603	592	580
1.00	*R*	216	227	239	251	263	275	287	299	313	327	341	355	370	386	403
	D	642	636	631	626	622	618	615	612	610	608	606	604	603	602	601
	e	600	620	636	647	654	657	659	658	656	651	644	635	625	614	602
1.10	*R*	197	208	219	230	241	252	263	275	288	301	314	328	342	357	373
	D	619	613	608	604	600	596	592	588	585	583	581	579	578	577	576
	e	605	628	646	659	668	673	676	676	673	667	659	650	640	629	616
1.20	*R*	180	190	200	210	221	232	243	255	267	279	292	305	319	334	349
	D	600	593	587	582	577	573	570	567	564	562	560	558	557	556	555
	e	606	631	650	663	674	678	681	681	678	673	667	660	650	639	627
1.30	*R*	167	176	186	196	206	216	226	236	247	259	271	283	296	310	325
	D	583	575	569	564	559	554	550	547	544	541	539	537	536	535	534
	e	606	635	654	666	674	679	682	683	681	676	669	661	652	642	632
1.40	*R*	155	164	174	183	192	202	212	221	231	242	253	264	276	290	305
	D	568	560	553	548	543	538	534	530	527	524	521	519	517	516	515
	e	603	636	658	671	678	684	683	682	679	675	670	664	656	647	636
1.50	*R*	145	154	163	172	181	190	199	209	219	229	239	250	262	275	289
	D	554	547	540	533	528	523	519	515	512	509	506	504	502	501	500
	e	594	630	654	670	679	683	683	681	678	673	667	660	652	643	633
1.60	*R*	136	144	153	162	171	180	189	198	207	216	226	237	249	261	274
	D	542	534	527	521	515	510	506	502	498	495	492	490	488	487	486
	e	580	617	643	659	669	675	677	676	673	669	663	656	648	639	630
1.80	*R*	122	130	138	146	154	162	170	179	188	197	207	218	229	240	252
	D	524	515	507	500	493	487	483	480	476	473	470	468	466	465	464
	e	560	602	626	642	652	657	659	659	657	654	649	642	634	626	617
2.00	*R*	110	118	126	134	142	149	157	165	173	182	191	201	212	223	234
	D	507	498	489	481	474	468	463	459	456	453	450	448	447	446	445
	e	548	590	615	630	638	642	644	643	641	637	632	626	618	610	601

THREE-BLADED PROPELLERS.

PROJECTED AREA RATIO = 0.54. THICKNESS RATIO = 0.03.

Point off one place for D, two for R, three for e.

Pitch Ratio.		Real Slip.														
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34
0.60	*R*	342	354	368	382	397	413	429	446	465	484	504	525			
	D	796	790	784	779	775	772	769	766	764	763	762	762			
	e	482	487	491	495	499	502	504	506	507	507	504	498			
0.65	*R*	317	329	342	355	369	385	401	419	437	457	477	497	517	538	
	D	773	766	759	753	749	745	742	740	739	738	737	736	735	735	
	e	492	497	502	506	509	512	514	516	517	517	515	512	506	498	
0.70	*R*	298	309	321	333	346	360	375	391	409	427	447	468	489	511	534
	D	754	746	739	733	727	723	719	716	714	712	711	711	711	712	712
	e	505	511	516	520	523	525	527	528	528	527	525	522	517	510	500
0.75	*R*	280	290	302	314	326	339	353	368	384	402	422	442	463	485	507
	D	735	727	720	713	708	703	699	696	693	691	690	689	688	688	689
	e	515	523	529	534	538	541	542	543	542	540	537	533	528	520	510
0.80	*R*	262	272	284	296	308	321	335	349	363	379	398	417	438	460	482
	D	717	709	702	696	690	685	681	677	674	671	669	667	666	666	667
	e	521	534	544	551	555	557	558	559	558	556	553	548	541	533	525
0.90	*R*	234	244	255	266	278	290	302	315	328	343	360	378	398	419	440
	D	684	677	671	665	659	654	649	645	641	638	635	633	632	631	630
	e	525	551	569	580	586	589	590	590	589	587	583	578	571	562	552
1.00	*R*	209	219	230	241	252	263	274	286	298	312	328	345	363	383	403
	D	653	646	639	633	628	623	619	616	613	610	607	604	602	600	600
	e	515	548	578	598	610	616	619	620	620	618	613	606	597	586	574
1.10	*R*	192	201	211	221	232	243	254	265	276	288	302	318	335	353	371
	D	628	621	614	608	602	597	593	590	587	584	581	579	577	575	573
	e	512	547	580	602	617	627	634	638	639	638	635	628	618	607	596
1.20	*R*	177	186	196	206	216	226	237	248	259	270	282	296	311	328	345
	D	606	598	590	584	578	574	570	567	564	562	560	558	556	554	552
	e	513	551	585	608	624	635	642	646	648	647	644	638	629	617	604
1.30	*R*	167	176	185	194	204	214	224	234	244	254	266	278	292	307	323
	D	587	579	571	564	558	553	549	546	543	541	539	537	535	533	532
	e	518	559	590	614	631	642	648	650	651	649	645	640	632	622	611
1.40	*R*	157	166	175	184	193	203	213	223	233	243	253	264	276	290	305
	D	570	561	553	546	540	535	531	527	524	522	520	518	517	516	515
	e	513	557	589	613	630	641	647	650	649	647	643	638	631	622	611
1.50	*R*	150	158	166	175	184	193	203	213	223	233	243	253	264	276	290
	D	555	546	538	531	525	520	515	511	508	506	504	503	502	501	500
	e	503	548	580	604	622	634	641	644	644	641	637	632	626	618	608
1.60	*R*	143	151	159	168	177	186	195	204	214	224	234	244	254	265	277
	D	542	533	525	518	512	506	501	497	494	491	489	488	487	486	485
	e	490	532	563	592	612	624	631	635	636	635	631	627	620	613	602
1.80	*R*	133	141	149	157	165	174	183	192	201	210	220	230	240	250	260
	D	518	509	501	494	488	482	477	473	470	468	466	464	463	462	462
	e	458	501	532	560	578	594	603	607	609	610	608	604	598	590	582
2.00	*R*	123	131	139	147	155	164	173	182	191	200	210	220	230	240	250
	D	494	485	477	470	464	459	454	451	448	446	445	444	444	444	444
	e	413	455	486	510	530	546	558	566	571	574	575	573	569	563	555

TWO-BLADED PROPELLERS.

PROJECTED AREA RATIO = 0.18. THICKNESS RATIO = 0.06.

Point off one place for *D*, two for *R*, three for *e*.

Pitch Ratio.		Real Slip.														
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34
0.60	*R*	399	413	428	443	457	472	486	501	516	532					
	D	731	731	732	734	736	739	741	743	746	749					
	e	554	555	554	552	548	543	537	531	524	517					
0.65	*R*	369	381	394	407	421	436	451	465	480	496	512				
	D	714	715	715	715	716	717	719	721	723	726	729				
	e	574	581	583	583	581	580	576	572	577	561	556				
0.70	*R*	342	353	365	377	390	404	418	434	450	466	484	502	521		
	D	697	697	697	698	698	699	700	702	704	706	708	712	716		
	e	596	600	603	605	604	602	599	596	592	588	583	578	572		
0.75	*R*	319	329	340	352	363	376	399	404	420	436	453	470	488	508	
	D	682	682	681	681	681	682	683	684	685	687	690	693	696	699	
	e	612	619	622	624	624	622	619	616	611	607	602	597	591	584	
0.80	*R*	298	309	320	330	342	354	366	379	392	406	421	437	456	476	497
	D	668	668	668	667	667	667	668	668	668	669	671	674	676	679	682
	e	628	634	638	640	640	639	637	634	631	627	621	615	608	600	591
0.90	*R*	263	273	283	293	304	314	325	337	349	362	376	390	406	423	440
	D	654	654	653	653	652	652	652	652	652	652	653	654	656	658	660
	e	651	658	661	663	664	663	662	660	657	653	648	642	635	627	619
1.00	*R*	236	245	254	263	273	284	295	306	317	328	341	354	367	382	397
	D	624	621	620	619	618	618	617	617	617	617	617	618	619	620	621
	e	682	686	689	691	692	691	689	686	682	677	671	663	655	646	637
1.10	*R*	216	222	230	239	248	258	268	279	290	301	313	325	338	352	366
	D	605	604	603	602	601	600	598	597	597	597	597	597	598	598	599
	e	697	703	708	712	713	712	709	704	699	693	686	679	671	662	651
1.20	*R*	196	203	210	217	226	236	246	256	266	277	288	300	313	326	339
	D	588	586	584	582	581	580	580	579	579	578	578	578	579	580	581
	e	708	716	722	726	728	726	725	721	716	709	700	691	682	672	661
1.30	*R*	182	188	194	201	209	217	226	236	246	256	267	279	291	303	317
	D	572	570	568	566	565	564	563	562	561	560	560	560	560	561	562
	e	720	727	731	734	736	736	735	732	727	719	711	701	691	680	669
1.40	*R*	170	175	180	187	195	203	212	221	230	240	250	261	273	285	297
	D	559	557	555	552	550	548	546	545	544	543	542	542	543	544	545
	e	731	736	740	743	745	745	743	740	735	728	719	708	698	688	677
1.50	*R*	153	160	167	173	179	186	193	201	209	218	227	237	248	259	271
	D	547	544	541	539	537	535	533	531	529	528	528	527	527	528	528
	e	737	744	749	752	753	752	750	746	740	732	722	713	703	692	681
1.60	*R*	146	152	158	163	168	175	182	189	197	205	214	224	235	247	258
	D	536	532	529	526	524	522	520	518	516	515	514	513	513	513	514
	e	738	746	752	756	757	757	756	752	746	737	728	718	707	694	682
1.80	*R*	130	135	141	147	153	159	166	173	181	189	198	207	216	226	236
	D	515	511	507	504	501	498	496	494	492	490	489	488	487	487	488
	e	728	738	746	752	756	757	756	752	745	736	725	714	703	692	681
2.00	*R*	115	120	126	132	138	145	151	158	166	173	181	189	198	207	216
	D	494	490	486	483	479	476	474	472	470	468	466	465	464	463	463
	e	702	718	729	737	742	745	744	742	738	731	723	714	704	692	680

AUXILIARY

R, revolutions; D, displacement,

R	$R^{\frac{2}{3}}$	$R^{\frac{1}{2}}$	S.H.P.	$\overline{\text{S.H.P.}}^{\frac{1}{6}}$	$\overline{\text{S.H.P.}}^{\frac{1}{4}}$	D	$D^{\frac{2}{3}}$
50	13.58	7.07	10	1.468	1.779	700	78.84
55	14.46	7.42	15	1.570	1.969	725	80.70
60	15.32	7.75	20	1.648	2.118	750	82.55
65	16.16	8.06	25	1.710	2.240	775	84.38
70	16.99	8.36	30	1.763	2.341	800	86.18
75	17.79	8.66	35	1.809	2.438	825	87.96
80	18.56	8.94	40	1.850	2.520	850	89.73
85	19.33	9.22	45	1.887	2.590	875	91.48
90	20.08	9.49	50	1.920	2.660	900	93.22
95	20.82	9.75	55	1.951	2.725	925	94.93
100	21.54	10.00	60	1.980	2.781	950	96.64
110	22.96	10.48	65	2.007	2.840	975	98.33
120	24.33	10.95	70	2.030	2.895	1,000	100.00
130	25.66	11.40	75	2.054	2.945	1,250	116.04
140	26.96	11.83	80	2.076	2.992	1,500	131.03
150	28.23	12.25	85	2.097	3.039	1,750	145.22
160	29.47	12.65	90	2.117	3.080	2,000	158.74
170	30.69	13.04	95	2.136	3.121	2,250	171.71
180	31.88	13.42	100	2.154	3.16	2,500	184.20
190	33.05	13.78	125	2.235	3.35	2,750	196.28
200	34.21	14.14	150	2.305	3.50	3,000	208.01
210	35.33	14.49	175	2.361	3.64	3,250	219.40
220	36.44	14.83	200	2.418	3.76	3,500	230.52
230	37.54	15.17	225	2.466	3.88	3,750	241.40
240	38.62	15.49	250	2.510	3.98	4,000	251.98
250	39.68	15.81	275	2.550	4.07	4,250	262.32
260	40.74	16.12	300	2.59	4.16	4,500	272.56
270	41.78	16.43	325	2.62	4.24	4,750	282.53
280	42.80	16.73	350	2.66	4.33	5,000	292.40
290	43.81	17.03	375	2.68	4.40	5,250	302.06
300	44.81	17.32	400	2.72	4.47	5,500	311.58
310	45.80	17.61	425	2.74	4.54	5,750	320.95
320	46.78	17.89	450	2.77	4.60	6,000	330.19
330	47.75	18.17	475	2.79	4.66	6,250	339.30
340	48.71	18.44	500	2.82	4.74	6,500	348.29
350	49.66	18.71	525	2.84	4.80	6,750	357.16
360	50.61	18.97	550	2.86	4.84	7,000	365.93
370	51.54	19.23	575	2.88	4.90	7,250	374.58
380	52.46	19.49	600	2.90	4.95	7,500	383.15
390	53.38	19.75	625	2.92	5.00	7,750	391.62
400	54.29	20.00	650	2.94	5.05	8,000	400.00
410	55.19	20.25	675	2.96	5.10	8,250	408.28
420	56.08	20.49	700	2.98	5.15	8,500	416.49
430	56.97	20.73	725	3.00	5.19	8,750	424.62
440	57.85	20.97	750	3.01	5.24	9,000	432.67
450	58.72	21.21	775	3.03	5.28	9,250	440.64
460	59.59	21.45	800	3.05	5.32	9,500	448.54
470	60.45	21.68	825	3.06	5.36	9,750	456.39
480	61.30	21.91	850	3.08	5.40	10,000	464.16
490	62.15	22.14	875	3.09	5.44	10,500	479.49

PROPELLER TABLE.

S.H.P., shaft horse-power.

R	$R^{\frac{2}{3}}$	$R^{\frac{1}{2}}$	S.H.P.	$\overline{\text{S.H.P.}}^{\frac{1}{6}}$	$\overline{\text{S.H.P.}}^{\frac{1}{4}}$	D	$D^{\frac{2}{3}}$
500	62.99	22.36	900	3.11	5.47	11,000	494.61
510	63.83	22.58	925	3.12	5.52	11,500	509.48
520	64.66	22.80	950	3.13	5.55	12,000	524.15
530	65.49	23.02	975	3.14	5.59	12,500	538.60
540	66.31	23.24	1,000	3.16	5.62	13,000	552.88
550	67.13	23.45	1,500	3.38	6.22	13,500	566.96
560	67.94	23.66	2,000	3.55	6.70	14,000	580.88
570	68.74	23.87	2,500	3.68	7.09	14,500	594.61
580	69.54	24.08	3,000	3.80	7.40	15,000	608.22
590	70.34	24.29	3,500	3.90	7.69	15,500	621.66
600	71.13	24.49	4,000	3.98	7.95	16,000	634.97
610	71.92	24.70	4,500	4.05	8.19	16,500	648.12
620	72.71	24.90	5,000	4.13	8.40	17,000	661.15
630	73.49	25.10	5,500	4.20	8.60	17,500	674.05
640	74.26	25.30	6,000	4.27	8.80	18,000	686.83
650	75.03	25.50	6,500	4.32	8.98	18,500	699.49
660	75.80	25.70	7,000	4.37	9.15	19,000	712.04
670	76.57	25.88	7,500	4.42	9.30	19,500	724.48
680	77.33	26.08	8,000	4.47	9.45	20,000	736.81
690	78.08	26.27	8,500	4.51	9.60	20,500	749.04
700	78.84	26.46	9,000	4.56	9.74	21,000	761.17
			9,500	4.60	9.87	21,500	773.20
			10,000	4.64	10.00	22,000	785.14
			10,500	4.67	10.13	22,500	796.99
			11,000	4.72	10.23	23,000	808.76
			11,500	4.75	10.35	23,500	820.44
			12,000	4.77	10.46	24,000	832.04
			12,500	4.82	10.58	24,500	843.55
			13,000	4.85	10.68	25,000	854.98
			13,500	4.87	10.78	25,500	866.35
			14,000	4.91	10.88	26,000	877.64
			14,500	4.94	10.98	26,500	888.86
			15,000	4.97	11.08	27,000	900.00
			15,500	4.99	11.18	27,500	911.08
			16,000	5.02	11.25	28,000	922.06
			16,500	5.05	11.32	28,500	933.04
			17,000	5.07	11.40	29,000	943.91
			17,500	5.10	11.50	29,500	954.73
			18,000	5.12	11.59	30,000	965.49
			18,500	5.14	11.65	30,500	976.18
			19,000	5.16	11.73	31,000	986.83
			19,500	5.19	11.81	31,500	997.40
			20,000	5.20	11.89	32,000	1007.9
			20,500	5.22	11.97	32,500	1018.5
			21,000	5.25	12.03	33,000	1028.8
			21,500	5.27	12.10	33,500	1039.3
			22,000	5.29	12.19	34,000	1049.7
			22,500	5.31	12.24	34,500	1059.8
			23,000	5.33	12.30	35,000	1070.3
			23,500	5.35	12.37		

POWERS OF

MODEL.

V	$V^{\frac{1}{6}}$	$V^{\frac{5}{4}}$	$V^{1.94}$	$V^{2.94}$	V^{3}	V^{5}
1.0	1.00	1.00	1.0	1.0	1.0	1.0
1.2	1.031	1.256	1.424	1.709	1.728	2.49
1.4	1.058	1.522	1.920	2.689	2.744	5.37
1.6	1.081	1.800	2.488	3.982	4.096	10.48
1.8	1.103	2.088	3.127	5.629	5.832	18.89
2.0	1.122	2.380	3.837	7.674	8.000	32.00
2.2	1.140	2.660	4.616	10.15	10.65	51.53
2.4	1.157	2.990	5.465	13.11	13.82	79.64
2.6	1.173	3.300	6.383	16.59	17.58	118.81
2.8	1.187	3.620	7.370	20.63	21.95	172.10
3.0	1.201	3.945	8.425	25.27	27.00	243.00
3.2	1.214	4.27	9.549	30.55	32.77	335.54
3.4	1.226	4.61	10.74	36.52	39.30	454.35
3.6	1.238	4.96	12.00	43.20	46.66	604.66
3.8	1.249	5.30	13.32	50.64	54.87	792.30
4.0	1.260	5.65	14.72	58.89	64.00	1,024.0
4.2	1.270	6.00	16.18	67.97	74.09	1,306.9
4.4	1.280	6.38	17.71	77.93	85.18	1,649.2
4.6	1.290	6.65	19.30	88.81	97.34	2,059.6
4.8	1.298	7.10	20.97	100.6	110.6	2,548.0
5.0	1.308	7.49	22.69	113.4	125.0	3,125.0
5.2	1.316	7.86	24.49	127.3	140.6	3,802.0
5.4	1.325	8.25	26.35	142.3	157.5	4,591.6
5.6	1.332	8.50	28.28	158.3	175.6	5,507.3
5.8	1.340	9.00	30.27	175.5	195.1	6,563.6
6.0	1.348	9.40	32.33	193.9	216.0	7,776.0
6.2	1.356	9.80	34.45	213.6	238.3	9,161.3
6.4	1.362	10.19	36.64	234.5	262.1	10,738
6.6	1.370	10.60	38.89	256.7	287.5	12,523
6.8	1.376	10.49	41.21	280.2	314.4	14,539
7.0	1.384	11.40	43.60	305.2	343.0	16,807
7.2	1.390	11.78	46.05	331.5	373.2	19,349
7.4	1.396	12.21	48.56	359.3	405.2	22,190
7.6	1.403	12.62	51.14	388.6	439.0	25,355
7.8	1.409	13.05	53.78	419.5	474.5	28,872
8.0	1.415	13.46	56.49	451.9	512.0	32,768
8.2	1.420	13.90	59.26	485.9	551.4	37,074
8.4	1.426	14.30	62.10	521.6	592.7	41,821
8.6	1.432	14.75	65.00	559.0	636.1	47,043
8.8	1.437	15.15	67.96	598.1	681.5	52,773
9.0	1.443	15.6	70.99	638.9	729.0	59,049
9.2	1.448	16.0	74.08	681.6	778.7	65,908
9.4	1.453	16.5	77.24	726.1	830.6	73,390
9.6	1.458	16.9	80.46	772.4	884.7	81,537
9.8	1.463	17.3	83.74	820.7	941.2	90,392
10.0	1.468	17.7	87.09	870.9	1000	100,000
10.5	1.480	18.9	95.74	1005	1158	127,630
11.0	1.492	20.0	104.7	1152	1331	161,050
11.5	1.503	21.2	114.2	1313	1521	201,140
12.0	1.513	22.3	124.0	1488	1728	248,830

SPEEDS.

SHIP.

V	$V^{\frac{1}{6}}$	$V^{\frac{5}{4}}$	$V^{1.825}$	$V^{2.825}$	V^3	V^5
1	1.0	1.0	1	1	1	1
2	1.122	2.380	3.54	7.09	8	32
3	1.201	3.945	7.42	22.2	27	243
4	1.260	5.650	12.55	50.2	64	1,024
5	1.308	7.49	18.86	94.3	125	3,125
6	1.348	9.40	26.31	157.8	216	7,776
7	1.384	11.40	34.85	244.0	343	16,807
8	1.415	13.46	44.47	355.8	512	32,768
9	1.443	15.60	55.14	496.2	729	59,049
10	1.468	17.70	66.83	668.3	1,000	100,000
11	1.492	20.00	79.53	874.8	1,331	161,050
12	1.513	22.38	93.21	1,118	1,728	248,830
13	1.533	24.68	107.8	1,402	2,197	371,290
14	1.553	27.07	123.5	1,729	2,744	537,820
15	1.570	29.51	140.0	2,101	3,375	759,370
16	1.587	32.00	157.5	2,521	4,096	1,048,600
17	1.605	34.51	176.0	2,992	4,913	1,419,900
18	1.620	37.08	195.3	3,516	5,832	1,889,600
19	1.634	39.67	215.6	4,097	6,859	2,476,100
20	1.648	42.30	236.8	4,735	8,000	3,200,000
21	1.661	44.96	258.8	5,435	9,261	4,084,100
22	1.675	47.64	281.7	6,199	10,648	5,153,600
23	1.686	50.37	305.6	7,028	12,167	6,436,300
24	1.698	53.12	330.2	7,926	13,824	7,962,600
25	1.710	55.90	355.8	8,895	15,625	9,765,600
26	1.721	58.71	382.2	9,938	17,576	11,881,000
27	1.732	61.55	409.4	11,056	19,683	14,349,000
28	1.742	64.40	437.5	12,252	21,952	17,210,000
29	1.753	67.30	466.5	13,529	24,389	20,511,000
30	1.762	70.21	496.3	14,889	27,000	24,300,000
31	1.771	73.15	526.9	16,334	29,791	28,629,000
32	1.780	76.11	558.3	17,867	32,768	33,554,000
33	1.790	79.09	590.5	19,490	35,937	39,135,000
34	1.800	82.10	623.6	21,204	39,304	45,435,000
35	1.810	85.13	657.5	23,014	42,875	52,522,000
36	1.819	88.18	692.2	24,920	46,656	60,466,000
37	1.826	91.25	727.7	26,925	50,653	69,344,000
38	1.834	94.34	764.0	29,033	54,872	79,235,000
39	1.841	97.46	801.1	23,243	59,319	90,224,000
40	1.849	100.65	839.0	33,560	64,000	102,400,000

LOGARITHMS.

Nat. Nos.	0	1	2	3	4	5	6	7	8	9	Proportional Parts.								
											1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7

LOGARITHMS.

Nat. Nos.	0	1	2	3	4	5	6	7	8	9	Proportional Parts.								
											1	2	3	4	5	6	7	8	9
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

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